ASRE STANDARDS

The
AMERICAN SOCIETY
OF REFRIGERATING
ENGINEERS

40 WEST 40 STREET
NEW YORK 18, NEW YORK

2A.

A.S.R.E. Circular No. 14—41

A.S.R.E. STANDARD METHODS OF RATING AND TESTING MECHANICAL CONDENSING UNITS

REVISED 1940

by

THE JOINT COMMITTEE

ON

RATING COMMERCIAL REFRIGERATING EQUIPMENT

Sponsored by The American Society of Refrigerating Engineers

Approved by A.S.R.E. Council, June 11, 1940

CENTRAL FOOD TECHNOLOGICAL RESEARCH INSTITUTE, * MYSORE. *
15 NOV [illegible]
LIBRARY

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40TH STREET . NEW YORK 18, N. Y.

PRICE 50 CENTS

RW 1624 ✓

Joint Committee on
Rating Commercial Refrigerating Equipment

This committee was organized in 1935 by The American Society of Refrigerating Engineers, jointly with the four other organized bodies in the following list of personnel, which includes men responsible for this standard (as of January 1940)

GLENN MUFFLY, Springfield, Ohio, *General Chairman*
CHESTER LICHTENBERG, Fort Wayne, Ind., *Secretary*

For The American Society of Refrigerating Engineers (Sponsor)—
Glenn Muffly, Chairman
R. H. Tull
Frank R. Zumbro

For the Refrigeration Division, National Electrical Manufacturers Association—
Chester Lichtenberg, Chairman
J. L. Gibson
L. A. Philipp

For the Refrigerating Machinery Association—
Louis S. Morse, Chairman
Willis H. Carrier

For the Air Conditioning Manufacturers Association—
Donald E. French, Chairman
J. L. Gibson
Louis S. Morse

For the American Society of Heating and Ventilating Engineers—
F. H. Faust, Chairman

This committee is also the author of standards for rating and testing Air Conditioning Equipment, Self-Contained Air Conditioning Units for Comfort Cooling, and Evaporative Condensers. Other projects in development includes codes for testing and rating Compressors and Condensers.

Reorganized in 1941, the committee now has three organization members, The American Society of Refrigerating Engineers, the American Society of Heating and Ventilating Engineers and the Air Conditioning and Refrigerating Machinery Association.

A.S.R.E. Standard Method of Rating Mechanical Condensing Units

Foreword

The capacity of a mechanical condensing unit depends upon its design and the temperature conditions under which it is applied and operated. This Standard cannot include all of the application variables which are encountered in practice. Therefore, conditions covering eight groups of application have been established for A. S. R. E. Standard rating purposes.

1. *A Mechanical Condensing Unit* is a specific refrigerating machine combination for a given refrigerant, consisting of a motor-driven compressor for operation at a given speed, a condenser, a liquid receiver and the regularly furnished accessories.

2. *The Capacity* of a mechanical condensing unit is the refrigerating effect in Btu. per hr. produced by the change in total heat content between the liquid refrigerant leaving the condensing unit per hour and the total heat content of the vapor refrigerant entering the condensing unit per hour.

3. *Each Capacity Rating* of a mechanical condensing unit shall be expressed in Btu. per hr. or in tons, each of 12,000 Btu. per hr., under conditions specified as follows:

a. Temperature of the ambient in deg. F.

b. Temperature of the saturated refrigerant vapor in deg. F.

c. Temperature of the refrigerant vapor entering the condensing unit in deg. F.

d. Temperature of the ingoing cooling water if any, in deg. F.

e. Temperature of the outgoing cooling water, if any, in deg. F.

f. "Wet bulb" temperature of air entering evaporative condenser, if any, in deg. F.

g. "Dry bulb" temperature of air entering air cooled condensers, if any, in deg. F.

4. *The A. S. R. E. Standard Capacity Rating* of a mechanical condensing unit shall be measured under conditions defined as follows:

4.1 The unit ratings shall be divided into standard groups based on saturated refrigerant vapor temperatures in the tabulation below:

This tabulation is part of Par. 4.1

Group No.	Saturated refrigerant vapor temperatures		*Water Cooled Condenser*			*Evaporatively Cooled Condenser*			*Air Cooled Condenser*	
			Cooling water temperature		*Refrigerant vapor temp. entering condensing unit*	*Air entering temperature*		*Refrigerant vapor temp. entering condensing unit*	*Air entering temp., Dry Bulb*	*Refrigerant vapor temp. entering condensing unit*
			In-going	*Out-going*		*Dry Bulb*	*Wet Bulb*			
I	Minus	10 F.	75 F.	85 F.	65 F.	90 F.	75 F.	65 F.	90 F.	80 F.
II	Plus	5 F.	75 F.	90 F.	65 F.	90 F.	75 F.	65 F.	90 F.	80 F.
III	Plus	20 F.	75 F.	90 F.	65 F.	90 F.	75 F.	65 F.	90 F.	80 F.
IV	Plus	40 F.	75 F.	95 F.	65 F.	90 F.	75 F.	65 F.	90 F.	80 F.
V	Minus	10 F.	75 F.	85 F.	15 F.	90 F.	75 F.	15 F.	90 F.	15 F
VI	Plus	5 F.	75 F.	90 F.	30 F.	90 F.	75 F.	30 F.	90 F.	30 F.
VII	Plus	20 F.	75 F.	90 F.	40 F.	90 F.	75 F.	40 F.	90 F.	40 F.
VIII	Plus	40 F.	75 F.	95 F.	55 F.	90 F.	75 F.	55 F.	90 F.	55 F.

The temperature of the refrigerant vapor entering a condensing unit depends primarily on the arrangement of the refrigerant circuit, the size of the condensing unit and the characteristics of the refrigerant.

Groups Nos. I, II, III and IV temperatures specified for the refrigerant vapor entering the condensing units are intended as a rating basis for condensing units which are usually applied with the refrigerant vapor entering the condensing unit at the specified temperatures.

Groups Nos. V, VI, VII, and VIII temperatures specified for the refrigerant vapor entering the condensing unit are intended as a rating basis for condensing units which cannot usually be applied with the refrigerant vapor entering the condensing unit at the temperatures specified in Groups Nos. I, II, III and IV.

(Concluded on page 4)

4.2 The saturated refrigerant vapor temperature shall be determined by the suction pressure as measured at the suction inlet connection to the condensing unit.

4.3 The standard ambient temperature for air cooled, air and water cooled or water cooled condensing units shall be 90°F. dry bulb. The standard ambient temperature for condensing units equipped with evaporative condensers shall be 90°F. dry bulb and 75°F. wet bulb.

5. *The Power Input Rating* of an electrically driven mechanical condensing unit is its total power input in watts when the unit is operated under the conditions listed as required in Section 3.

6. *The A. S. R. E. Standard Power Input Rating* of an electrically driven mechanical condensing unit is its total input in watts when the unit is operated under a given group of the conditions specified in Section 4.

7. *The Performance Factor* of a mechanical condensing unit is the ratio of its capacity to its energy input, expressed in Btu. per watt-hr. or in tons, each of 12,000 Btu. per hr., per kilowatt.

8. *The Cooling Water Consumption* of a mechanical condensing unit is the total number of gallons per hour required under the conditions listed as required in section 3.

9. *The A. S. R. E. Standard Cooling Water Consumption* of a mechanical condensing unit is the total number of gallons per hour required under a given group of the conditions specified in section 4.

10. *Published Capacity Ratings* shall include one or more A. S. R. E. Standard Capacity Ratings, identified as such by Group Numbers, each with its associated compressor speed, Power Input Rating and, if water is used, the Cooling Water Consumption.

Methods for Testing are given on following pages.

A.S.R.E. Standard Methods for Testing Mechanical Condensing Units

11. *Standard Test Methods* shall be used in determining the standard A. S. R. E. rating of mechanical condensing units. There shall be a primary test and a simultaneous confirming test. The capacity determined by the confirming test shall agree with that determined by the primary test within 3 per cent but the results of the primary test shall govern for rating purposes.

11.1 Primary Tests are as follows:

a. Brine tank calorimeter. (Sec. 19.)
b. Concentric tube calorimeter. (Sec. 20.)
c. Secondary refrigerant calorimeter. (Sec. 21.)
d. Flooded system primary refrigerant calorimeter. (Sec. 22.)
e. Dry system primary refrigerant calorimeter. (Sec. 23.)

11.2 Confirming Tests are as follows:

a. Liquid refrigerant quantity method. (Sec. 24.)
b. Liquid refrigerant quantity meter and flow meter. (Sec. 25.)
c. Condenser water method. (Sec. 26.)

12. *Values* to be determined under the conditions specified in the A.S.R.E. Standard Method of Rating Condensing Units are:

a. Refrigerating capacity in Btu. per hr. or tons, each of 12,000 Btu. per hr.
b. Power input in watts or kilowatts.
c. Cooling water consumption in gal. per hr.

13. *Types of Capacity Measuring Apparatus*

13.1 A Condensing Unit Calorimeter is an apparatus for determining the refrigerating capacity of a condensing unit by measuring the heat input required to balance the refrigerating effect produced in the evaporator by the condensing unit.

13.11 A brine tank calorimeter includes an evaporator and a heating unit submerged in an externally insulated brine bath. The refrigerating effect produced in the evaporator by the condensing unit is balanced by heat added to the brine bath.

13.12 A concentric tube calorimeter includes a heat exchanger for the transfer of heat from a fluid heating medium to a vaporizing refrigerant. The refrigerating effect produced by the condensing unit is balanced by heat given up by the fluid heating medium.

13.13 A secondary refrigerant calorimeter includes an evaporator located in the top and a heating unit located in the bottom of a pressure-tight container which is partially filled with a secondary refrigerant. The refrigerating effect produced in the evaporator by the condensing unit is balanced by heat added to the secondary refrigerant.

13.14 A primary refrigerant calorimeter includes an evaporator with a heating unit in direct thermal contact with a primary refrigerant. The refrigerating effect produced in the evaporator by the condensing unit is balanced by heat added to the primary refrigerant.

13.2 A Liquid Refrigerant Quantity Meter is an apparatus for determining the refrigerating capacity of a condensing unit by measuring the circulation of the refrigerant. It includes hand operated measuring cylinders or an integrating mechanism and supplementary devices for determining the quantity of liquid refrigerant circulated in a given period of time. The refrigerating effect produced by the condensing unit is the product of the pounds of refrigerant circulated per hour and the difference between the heat content per pound of refrigerant vapor entering and refrigerant liquid leaving the condensing unit.

13.3 A liquid Refrigerant Flow Meter includes a mechanism and supplementary devices for determining the instantaneous rate of circulation of liquid refrigerant. The refrigerating effect produced by the condensing unit is the product of the pounds of refrigerant liquid circulated per hour and the difference between the heat content per pound of refrigerant vapor entering and refrigerant leaving the condensing unit.

13.4 Condenser Water Method determines the refrigerating effect produced by the condensing unit from the heat transferred to the condenser water.

14. *Instruments*, whose types and accuracies are listed below, shall be calibrated against secondary standards before and after each test.

14.1 Temperature measuring instruments.

14.11 Temperature measurements shall be made with one or more of the following instruments:

a. Mercury-in-glass thermometers.
b. Thermocouples.
c. Electric resistance thermometers.

14.12 Accuracy shall be within the following limits:

a. Brine or water in calorimeters, ± 0.1°F.
b. Water in condensers, when data are used in confirming tests, ± 0.1°F.
c. "Wet bulb" temperatures, ± 0.1°F.
d. All other temperatures, ± 0.5°F.

14.2 Pressure measuring instruments.

14.21 Pressure measurements shall be made with one or more of the following instruments:

a. Mercury column.
b. Bourdon tube gage.

14.22 Accuracy shall be within the following limits:

a. Suction pressure, absolute, ± 1 per cent.
b. Other pressures, absolute, ± 2 per cent.

14.3 Electrical measuring instruments.

14.31 Electrical measurements shall be made with one or more of the following instruments:

a. Indicating.
b. Integrating.

14.32 Accuracy shall be within the following limits

a. Indicating instruments within 0.5 per cent of full scale reading

b. Integrating instruments within 1 per cent.

14.4 Refrigerant flow measuring instruments.

14.41 Volume measurements shall be made with one or more of the following instruments:

a. Liquid refrigerant quantity meter.

b. Liquid refrigerant flow meter.

14.42 Accuracy shall be within ± 1 per cent.

14.5 Brine or water flow measuring instruments.

14.51 Volume measurements shall be made with one or more of the following instruments:

a. Liquid quantity meter, measuring either weight or volume.

b. Liquid flow meter.

14.52 Accuracy shall be within ± 1 per cent.

14.6 Speed measuring instruments.

14.61 Speed measurements shall be made with one or more of the following instruments:

a. Revolution counter.

b. Tachometer.

c. Stroboscope.

d. Oscillograph.

14.62 Accuracy shall be within ± 0.25 per cent.

14.7 Time measurements shall be made with apparatus whose accuracy is within ± 0.1 per cent.

14.8 Weight measurements shall be made with apparatus whose accuracy is within ± 0.2 per cent.

15. *Tables of Physical Properties.* Physical data used in this test procedure shall be obtained from International Critical Tables or the current issue of *Refrigerating Data Book.*

16. *Tests*

16.1 Preparation.

16.11 The condensing unit shall have been operated for a total of not less than 72 hr. before capacity test is started.

16.12 The condensing unit shall be placed in a room in which the temperature can be controlled.

16.13 Provision shall be made to insure that no refrigerant vapor leaves the condensing unit.

16.14 The correct oil and refrigerant charges shall be placed in the proper locations in the refrigerating system and the refrigerant shall be maintained at normal operating level in the condensing unit liquid receiver, if the unit has one.

16.15 The condensing unit shall be operated for at least 1½ hr. and then shut down with the suction stop valve and the condensing unit liquid outlet valve closed. The temperature corresponding to the saturated static vapor pressure on the high pressure side of the condensing unit shall be within 2°F. of room temperature after the unit has been allowed to stand idle in a constant temperature room for at least 12 hr.

16.16 Mechanical checks shall be made to insure refrigerating system gas tightness, correct belt tension, compressor valve tightness, bearing freeness, correct compressor rotation and condensing unit mounting comparable to service conditions.

16.2 Operation and Limits.

16.21 Start the compressor and regulate the voltage at the motor terminals to within ± 2 per cent of motor rated voltage.

16.22 Adjust low pressure side conditions to specified suction pressure at condensing unit ± 1 per cent of absolute pressure.

16.23 Adjust the temperature of the suction vapor entering the condensing unit to a temperature specified in Sec. 4 of the A. S. R. E. Standard Method of Rating Mechanical Condensing Units (pages 3 and 4 hereof) plus or minus 5°F., excepting that where the capacity measuring apparatus does not permit adjusting the temperature of the vapor entering the condensing unit to a temperature specified in Sec. 4 the test may be made at a temperature which can be obtained if a desuperheater is added to cool the suction line and thereby obtain the specified temperature for the refrigerant vapor entering the condensing unit, with the capacity rating corrected to A. S. R. E. Standard Capacity Rating conditions in accordance with the formula in Par. 19.41, 20.41, 21.41, 22.41, 23.41, 24.41, 25.41 or 26.41, depending upon the type of capacity measuring apparatus used.

16.24 Water cooled condensing units. Adjust the temperature of the inlet water flowing to the condensing unit to 75°F., and adjust the water rate to give an outlet temperature indicated in Sec. 4 of A. S. R. E. Standard Method of Rating Condensing Units. The ambient temperature at the condensing unit shall be taken as the average reading of four temperature measuring instruments placed in a horizontal plane midway of the height of the condensing unit and eighteen inches from each side of the unit. The average ambient temperature shall be 90°F. ± 2°F. No abnormal air flow shall be created over the condensing unit.

16.25 Air-cooled condensing units. The temperature of the air flowing to the condensing unit shall be taken as the average reading of at least four temperature measuring instruments so located as to indicate the average air temperature. Each temperature measuring instrument shall be protected against radiant energy so that true "dry bulb" temperatures may be read. The temperature of the air flowing to the condensing unit shall be adjusted to 90°F. ± 1.0°F. No abnormal air flow shall be created over the condensing unit.

16.26 Evaporatively-cooled condensing units. The "wet bulb" temperature of the air entering the evaporative condenser of the condensing unit shall be held at 75°F. ± 0.5°F. The measurement of "wet bulb" temperature shall be made with a sling or aspirating psychrometer unless the inlet air velocity exceeds 1000 ft. per min., in which case "wet bulb" temperature measuring instruments may be paired and located at the centers of equal areas of one square foot. These instruments shall be shielded if exposed to radiant heat. The sling or aspirating psychrometer shall be used in such a manner that the temperature readings represent the average values of the samples of the entering air. The ambient temperature at the condensing unit shall be taken as the average reading of four temperature measuring instruments placed in a horizontal plane midway of the height of the condensing unit and eighteen inches from each side of the unit. The average ambient temperature shall be 90°F. ± 2°F. No abnormal air flow shall be created over the condensing unit.

16.27 Operate the condensing unit, before taking readings, until steady temperatures and mechanical equilibrium are established.

17. *Condensing Unit Data to be Recorded* shall include

a. Pressure of refrigerant vapor entering the unit.

b. Temperature of refrigerant vapor entering the unit, measured at 24 in. from the compressor.

c. Condenser pressure at inlet.

d. Temperature of liquid refrigerant leaving the unit.

e. Ambient temperature at unit.

f. Temperature of cooling water entering the unit.

g. Temperature of cooling water leaving the unit.

h. Gallons of cooling water per hour.

i. Pressure of cooling water entering the unit.

j. Pressure of cooling water leaving the unit.

k. Revolutions per minute of compressor.

l. Revolutions per minute of motor.

m. Watts input to condensing unit.

n. Volts at motor terminals.

o. Frequency at motor terminals.

p. Barometric pressure.

q. Entering air "wet bulb" temperature.

18. *Report of Test* shall include

a. Test number.

b. Date of test.

c. Condensing unit manufacturer's name.

d. Condensing unit model designation.

e. Motor nameplate rating.

f. Compressor test speed.

g. Compressor displacement in cubic inches per revolution.

h. Refrigerant name.

i. A. S. R. E. rating group.

j. Condensing unit capacity in Btu. per hr. or in tons, each of 12,000 Btu. per hr.

k. Motor input in watts or kilowatts.

l. Water consumption in gal. per hr. as corrected to the specified temperature rise.

m. List of condensing unit accessories regularly furnished and in place during the test.

19. *Brine Tank Calorimeter Method*

19.1 Description.

19.11 The evaporator shall be of the dry system coil type with sufficient surface to superheat the refrigerant vapor leaving the calorimeter to approximately 65°F. with a brine temperature not exceeding 100°F. and yet being compact so that a minimum amount of brine will be required.

19.12 The refrigerant control may be of either the hand regulated or the automatic expansion valve type and shall be located close to the brine tank. The control and the refrigerant line to the brine tank shall be insulated to minimize loss of refrigeration.

19.13 The heaters shall be of the electric immersion type with ample margin in capacity to provide ready control of the brine temperature.

19.14 The brine bath shall have a low enough freezing point to prevent it congealing at a temperature corresponding to the operating suction pressure.

19.15 Stirrers shall be provided for agitating the brine. They shall have sufficient capacity so that the maximum variation in temperature throughout the bath will not exceed 5°F. The heat added to the brine by agitation shall be determined.

19.16 The tank shall be insulated in such a manner that the heat leakage will not be greater than 5 per cent of the capacity of the condensing unit.

19.2 Calibration of the brine tank shall be performed by the heat absorption method. This shall be carried out as follows:

19.21 Four temperature measuring instruments shall be used to indicate the ambient temperature around the calorimeter, each instrument being located approximately in the middle of one of the vertical outside wall areas of the calorimeter, and spaced ½ in. from such wall. During calibration, the average ambient temperature indicated by the average of the four temperature readings shall be held constant within ± 2°F. at any desired value not exceeding 90°F.

19.22 Stirrer motors shall be operated and calorimeter heater input adjusted to maintain average brine temperature at approximately 25°F. above the ambient temperature.

19.23 After thermal equilibrium has been established the heat shall be held constant, and ambient and brine temperatures shall be recorded each hour for not less than 10 hr.

19.24 The mean temperature difference between brine and ambient shall be determined from the difference between the average of brine temperatures and average of ambient temperatures during the period of thermal equilibrium.

19.25 The heat leakage coefficient is given by:

$$q_l = \frac{Q_h + Q_s}{t_b - t_a}$$

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

Q_h = calorimeter heater input in Btu. per hr.

Q_s = calorimeter stirrer heat added to the brine in Btu. per hr.

t_a = average ambient temperature around the calorimeter in °F.

t_b = average brine temperature in °F.

19.26 The heat leakage is given by:

$$Q_l = q_l\ (t_a - t_b)$$

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

Q_l = heat leakage into calorimeter in Btu. per hr.

t_a = average ambient temperature in °F.

t_b = average brine temperature in °F.

19.27 This method is recommended, but the calorimeter may be calibrated for heat leakage by any other equivalent method.

19.3 Procedure.

19.31 The specified pressure of refrigerant vapor entering the condensing unit is adjusted by means of the refrigerant control, and the specified temperature of the refrigerant vapor entering the condensing unit is adjusted by varying the heat input to the brine bath.

19.32 Calorimeter data to be recorded after condensing unit rating specified conditions have become established are:

a. Pressure of refrigerant vapor at evaporator outlet.
b. Temperature of refrigerant vapor at evaporator outlet.
c. Pressure of liquid refrigerant entering refrigerant control.
d. Temperature of liquid refrigerant entering refrigerant control.
e. Temperature of brine.
f. Ambient temperature at calorimeter.
g. Calorimeter heater input.
h. Calorimeter stirrer heat input.

19.33 All readings shall be taken every 30 min.

19.34 Test shall be continued until four successive readings are within the allowable limits given in Section 16 and variation in brine temperature between the beginning and end of the test is not great enough to cause an error of more than 1 percent in condensing unit capacity.

19.4 Calculators.

19.41 The capacity of the condensing unit shall be determined from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}} (Q_h + Q_s + Q_l)$$

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit in Btu. per lb.

h_{f2} = heat content of the refrigerant liquid entering the refrigerant control in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit under conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units in Btu. per lb.

h_{g2} = Heat content of the refrigerant vapor leaving the calorimeter in Btu. per lb.

Q = condensing unit capacity, in Btu. per hr.

Q_h = calorimeter heater input, in Btu. per hr.

Q_s = calorimeter stirrer heat input, in Btu. per hr.

Q_l = calorimeter heat leakage gain, in Btu. per hr.

20. *Concentric Tube Calorimeter Method*

20.1 Description.

20.11 A concentric tube calorimeter consists of two independent concentric fluid circuits which are located in heat exchange relationship to each other. Liquid refrigerant is introduced into the inner circuit where it is vaporized and the vapor superheated as provided in Sec. 4. A heating medium of known properties is circulated through the outer circuit where it supplies the heat required to vaporize and superheat the refrigerant.

20.12 The refrigerant control may be of either the hand regulated or the automatic expansion valve type and shall be located close to the calorimeter. The control and the refrigerant line to the calorimeter shall be insulated to minimize loss of refrigeration.

20.13 The calorimeter shall be insulated in such a manner that the heat leakage will not be greater than 5 per cent of the capacity of the condensing unit.

20.2 Calibration.

20.21. The heat leakage of the calorimeter shall be determined by circulating the heating medium through the outer circuit of the calorimeter which is subject to no load other than heat leakage.

a. If water or brine is used, the temperature difference between the water or brine into and out of the calorimeter shall be not less than 10°F. and shall be established by regulating the rate of flow and the inlet temperature of the water or brine. The ambient temperature during this calibration shall be held constant within ± 2°F. at any desired value not exceeding 90°F. The inlet temperature of the water or brine shall be at least 30°F. above the ambient temperature. The test shall be continued until four successive brine or water temperatures taken at one hour intervals at both water or brine inlet and outlet of the calorimeter and with constant rate of flow show fluctuations no greater than ± 0.2°F.

b. If steam is used as the heating medium, the heat leakage of the calorimeter shall be determined by collecting and weighing the steam condensate from the heating medium circuit. The steam pressure shall be held constant within ± 0.5 lb. per sq. in. at any desired value. The ambient temperature during this calibration shall be held constant within ± 2°F. at any desired value not exceeding 90°F. Precautions must be taken to maintain the steam surface fully active without air or water binding, to maintain not less than 8°F. superheat on the steam admitted to the calorimeter, and to prevent loss by evaporation of condensate in the collecting vessel either by subcooling the condensate or adequately covering the vessel. The average outer surface temperature of the concentric tube shall be determined by at least ten equally spaced temperature measuring devices. The test shall be continued until four successive readings of condensate weight taken at not less than one hour intervals are within ± 10 per cent of each other.

20.22 The heat leakage coefficient is given by:

(a) If water or brine is used:

$$q_l = \frac{WC\,(t_1 - t_2)}{\frac{1}{2}\,(t_2 + t_1) - t_a}$$

C = specific heat of water or brine.

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

t_1 = temperature of water or brine entering the calorimeter, in °F.

t_2 = temperature of water or brine leaving the calorimeter, in °F.

t_a = average ambient temperature around the calorimeter, in °F.

W = weight of water or brine circulated in lb. per hr.

(b) If steam is used:

$$q_l = \frac{(h_{s1} - h_{s2})\,W_s}{t_c - t_a}$$

h_{s1} = heat content of entering steam in Btu. per lb.

h_{s2} = heat content of steam condensate at leaving temperature in Btu. per lb.

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter in °F.

t_c = concentric tube average outer surface temperature in °F.

W_s = weight of steam condensate in lb. per hr.

20.23 The heat leakage is given by:

(a) If water or brine is used:

$$Q_l = q_l\,[\tfrac{1}{2}\,(t_2 + t_1) - t_a]$$

Q_l = heat leakage from calorimeter in Btu. per hr.

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter in °F.

t_1 = temperature of water or brine entering the calorimeter in °F.

t_2 = temperature of water or brine leaving the calorimeter in °F.

(b) If steam is used:

$$Q_l = q_l\,(t_c - t_a)$$

Q_l = heat leakage from calorimeter in Btu. per hr.

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter in °F.

t_c = concentric tube outer surface average temperature, in °F.

20.24 This method is recommended, but the calorimeter may be calibrated for heat leakage by any other equivalent method.

20.3 Procedure.

20.31 The specified suction pressure at the condensing unit is adjusted by means of the refrigerant control and the specified temperature of the suction vapor at the condensing unit is adjusted by varying the heat input to the calorimeter.

20.32 If water or brine is used, the temperatures of the water or brine entering and leaving the calorimeter during the test shall be maintained to within ± 0.2°F. The weight of water or brine circulated through the calorimeter per hour shall be maintained within an accuracy of ± 1 per cent.

20.33 If steam is used as the heating medium, provision must be made to insure that only dry steam enters the calorimeter by maintaining not less than 8°F. superheat. The steam pressure in the calorimeter shall be measured with a mercury column to an accuracy of ± 2 per cent, and the temperature of entering steam and leaving condensate shall be read to an accuracy of ± 0.5°F. The weight of condensate shall be measured to an accuracy of ± 1 per cent, and precautions shall be taken to prevent loss of condensate by evaporation. Measurement of the surface temperature of the steam pipe shall be made at not less than ten equally spaced points.

20.34 The calorimeter data to be recorded, after condensing unit rating specified conditions have become established, are:

a. Pressure of refrigerant vapor at evaporator outlet.

b. Temperature of refrigerant vapor at evaporator outlet.

c. Pressure of liquid refrigerant before refrigerant control.

d. Temperature of liquid refrigerant before refrigerant control.

e. Ambient temperature at calorimeter.

f. Temperature of water or brine entering calorimeter, when water or brine is used.

g. Temperature of water or brine leaving calorimeter, when water or brine is used.

h. Weight of brine or water circulated per hr., when water or brine is used.

i. Temperature of steam entering calorimeter, when steam is used.

j. Pressure of steam in calorimeter, when steam is used.

k. Temperature of condensate leaving calorimeter, when steam is used.

l. Weight of condensate in pounds per hour, when steam is used.

m. Surface temperatures of steam pipe, when steam is used.

20.35 All readings shall be taken every 20 min.

20.36 Test shall be continued until four successive readings are within the allowable limits given in Section 16 and the variations in temperature and pressure conditions of the heating medium during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

20.4 Calculations.

20.41 The capacity of the condensing unit shall be determined from the following:

a. When water or brine is used:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}} \left[WC\ (t_1 - t_2) - Q_l \right]$$

C = specific heat of water or brine.

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit, in Btu. per lb.

h_{f2} = heat content of the refrigerant liquid entering the refrigerant control, in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit, under conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

h_{g2} = heat content of the refrigerant vapor leaving the calorimeter, in Btu. per lb.

Q = condensing unit capacity, in Btu. per hr.

Q_l = calorimeter heat leakage loss to the ambient atmosphere, in Btu. per hr.

t_1 = temperature of the water or brine entering the calorimeter, in °F.

t_2 = temperature of the water or brine leaving the calorimeter, in °F.

W = weight of water or brine circulated, in lb. per hr.

b. When steam is used:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}} \left[W_s\ (h_{s1} - h_{s2}) - Q_l \right]$$

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit, in Btu. per lb.

h_{f2} = heat content of the refrigerant liquid entering the refrigerant control, in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit under conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

h_{g2} = heat content of the refrigerant vapor leaving the calorimeter, in Btu. per lb.

h_{s1} = heat content of entering steam, in Btu. per lb.

h_{s2} = heat content of steam condensate at leaving temperature, in Btu. per lb.

Q_l = calorimeter heat leakage loss to the ambient atmosphere, in Btu. per hr.

W_s = weight of steam condensate, in lb. per hr.

21. *Secondary Refrigerant Calorimeter Method*

21.1 Description.

21.11 The secondary refrigerant calorimeter shall consist of a direct expansion coil or series of coils in parallel serving as a primary evaporator. This evaporator is suspended in the upper part of a pressure-tight vessel. A secondary refrigerant is placed in the bottom of this vessel and arrangements are made to supply heat to the liquid.

21.12 The refrigerant control may be of either the hand regulated or the automatic expansion valve type and should be located close to the calorimeter. The control and the refrigerant line to the calorimeter should be insulated to minimize loss of refrigeration.

21.13 The calorimeter shall be insulated in such a manner that the heat leakage will not be greater than 5 per cent of the capacity of the condensing unit.

21.14 Means shall be provided to obtain the pressure of the calorimeter secondary refrigerant. The gage used for this purpose shall be such that it can be accurately read to within 0.5 lb. per sq. in.

21.15 The calorimeter shall be equipped with a safety switch which will stop the flow of heat into the secondary refrigerant at a predetermined pressure.

21.2 Calibration.

21.21 The heat leakage of the calorimeter shall be determined by adjusting the heat input to the secondary refrigerant to a rate sufficient to maintain the secondary refrigerant at a pressure corresponding to approximately 25°F. above the ambient. The ambient temperature shall be held constant within ± 2°F. at any desired value not exceeding 90°F. The heat input to the secondary refrigerant shall be maintained constant, and pressure readings shall be taken at one hour intervals and continued until four successive values of saturated temperature do not differ by more than 1°F.

21.22 The heat leakage coefficient is given by:

$$q_l = \frac{Q_h}{t_s - t_a}$$

q_l = calorimeter heat leakage coefficient, in Btu. per hr. per °F. difference.

Q_h = calorimeter heat input, in Btu. per hr.

t_a = ambient temperature, in °F.

t_s = saturated secondary refrigerant temperature, in °F.

21.23 The heat leakage is given by:

$$Q_l = q_l\ (t_a - t_s)$$

Q_l = heat leakage into calorimeter, in Btu. per hr.

q_l = heat leakage coefficient, in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter, in °F.

t_s = saturated secondary refrigerant temperature, in °F.

21.24 This method is recommended, but the calorimeter may be calibrated for heat leakage by any other equivalent method.

21.3 Procedure.

21.31 The specified suction pressure at the condensing unit is adjusted by means of the refrigerant control, and the specified temperature of the suction vapor at the condensing unit is adjusted by varying the heat input to the secondary refrigerant.

21.32 The calorimeter data to be recorded, after the condensing unit rating specified conditions have become established, are:

a. Pressure of refrigerant vapor at evaporator outlet.

b. Temperature of refrigerant vapor at evaporator outlet.

c. Pressure of liquid refrigerant entering refrigerant control.

d. Temperature of liquid refrigerant entering refrigerant control.

e. Ambient temperature at calorimeter.

f. Pressure of secondary refrigerant.

g. Heat input to secondary refrigerant.

21.33 All readings shall be taken every 20 min.

21.34 Tests shall be continued until four successive readings are within the allowable limits given in Section 16 and the variation in heat input to the calorimeter during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

21.4 Calculations.

21.41 The capacity of the condensing unit shall be determined from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}} (Q_h + Q_l)$$

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit, in Btu. per lb.

h_{f2} = heat content of the refrigerant liquid entering the refrigerant control, in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit, under conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

h_{g2} = heat content of the refrigerant vapor leaving the calorimeter, in Btu. per lb.

Q = condensing unit capacity, in Btu. per hr.

Q_h = calorimeter heater input, in Btu. per hr.

Q_l = heat leakage into calorimeter, in Btu. per hr.

22. *Flooded System Primary Refrigerant Calorimeter Method*

22.1 Description.

22.11 The flooded system primary refrigerant calorimeter shall consist of a pressure-tight vessel or vessels in parallel, containing liquid refrigerant to which heat is supplied by heating elements in direct thermal contact with the liquid refrigerant.

22.12 The refrigerant control may be of either the hand regulated or the automatic expansion valve type and should be located close to the calorimeter. The control and the refrigerant line to the calorimeter should be insulated to minimize loss of refrigeration.

22.13 The calorimeter shall be insulated in such a manner that the heat leakage will not exceed 5 per cent of the capacity of the condensing unit.

22.2 Calibration of the calorimeter may be performed by either the heat absorption method or the small condensing unit method. These shall be carried out as follows:

22.21 Heat absorption method:

a. The calorimeter shall be filled with liquid refrigerant to its normal operating level and the liquid inlet and vapor outlet stop valves shall be closed. Holding the ambient temperature constant within ± 2°F. at any desired value not exceeding 90°F., supply enough heat to maintain a temperature approximately 25°F. above the ambient. The heat input to the refrigerant shall be maintained constant, and pressure readings shall be taken at one hour intervals and continued until four successive values of saturated temperature do not differ by more than 1°F.

b. The heat leakage coefficient is given by:

$$q_l = \frac{Q_h}{t_r - t_a}$$

q_l = calorimeter heat leakage coefficient, in Btu. per hr. per °F. difference.

Q_h = calorimeter heat input, in Btu. per hr.

t_a = average ambient temperature around the calorimeter, in °F.

t_r = saturated refrigerant temperature, in °F.

22.22 Small condensing unit method.

a. With the calorimeter in a 90°F. ambient and no load applied other than heat leakage, a small condensing unit is operated until steady conditions are reached. Then the condensed liquid refrigerant is collected and measured in volume measuring vessels by the method of Section 24 over a period of time sufficient to accumulate not less than a 6-in height of liquid in the measuring vessel.

b. The heat leakage coefficient is given by:

$$q_l = \frac{Vd\,(h_{g2} - h_{f2})}{t_a - t_r}$$

d = density of liquid refrigerant, for the temperature at which flow is measured, in pounds per cubic foot.

h_{f2} = heat content of refrigerant liquid entering the refrigerant control, in Btu. per lb.

h_{g2} = heat content of the refrigerant vapor leaving the calorimeter, in Btu. per lb.

q_l = calorimeter heat leakage coefficient, in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter, in °F.

t_r = saturated refrigerant temperature, in °F.

V = flow of liquid refrigerant, in cu. ft. per hr.

22.23 The heat leakage is given by:

$$Q_l = q_l\ (t_a - t_r)$$

Q_l = heat leakage into calorimeter, in Btu. per hr.

q_l = heat leakage coefficient, in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter, in °F.

t_r = saturated refrigerant temperature, in °F.

22.24 These methods are recommended, but the calorimeter may be calibrated for heat leakage by any other equivalent method.

22.3 Procedure.

22.31 The Specified suction pressure at the condensing unit is adjusted by means of the refrigerant control, and the specified temperature of the suction vapor at the condensing unit is adjusted by varying the heat input to the calorimeter.

22.32 Calorimeter data to be recorded, after the condensing unit rating specified conditions have become established, are:

a. Pressure of refrigerant vapor at evaporator outlet.
b. Temperature of refrigerant vapor at evaporator outlet.
c. Pressure of liquid refrigerant entering refrigerant control.
d. Temperature of liquid refrigerant entering refrigerant control.
e. Ambient temperature at calorimeter.
f. Calorimeter heater input.

22.33 All readings shall be taken every 20 min.

22.34 Test shall be continued until four successive readings are within the allowable limits given in Section 16 and the variation in heat input to the calorimeter during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

22.4 Calculations.

22.41 The capacity of the condensing unit shall be determined from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\ (Q_h + Q_l)$$

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit, in Btu. per lb.

h_{f2} = heat content of the refrigerant liquid entering the refrigerant control, in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

h_{g2} = heat content of the refrigerant vapor leaving the calorimeter, in Btu. per lb.

Q = condensing unit capacity, in Btu. per hr.

Q_h = calorimeter heater input, in Btu. per hr.

Q_l = heat leakage into calorimeter, in Btu. per hr.

23. *Dry System Primary Refrigerant Calorimeter Method.*

23.1 Description.

23.11 The dry system primary refrigerant calorimeter is similar to the system described in Sec. 22 except that a dry type evaporator is used and the necessary modifications in procedure are made to develop the full accuracy of this method.

23.12 The dry system primary refrigerant calorimeter shall consist of a series of pressure-tight tubular vessels of suitable length and diameter for accomplishing the evaporation of the refrigerant circulated by the condensing unit on test. The tubular vessels shall be electrically isolated and equipped with electrical heating means such as; tube-type electric heaters within the tubular vessels, wire-type heaters wound about the tubular vessels, heaters thermally well bonded to the tubular vessels, or the heating may be accomplished by passing current through the tubular vessels themselves, utilizing the heat thus generated to evaporate the refrigerant.

23.13 The refrigerant control may be of either the hand regulated or the automatic expansion valve type and should be located close to the calorimeter. The control and the refrigerant line to the calorimeter shall be insulated to minimize loss of refrigeration.

23.14 The calorimeter shall be insulated in such a manner that the heat leakage does not exceed 5 per cent of the capacity of the condensing unit.

23.15 The surfaces of the pressure vessels, and of the electrical insulation of the heaters, if they are external to the pressure vessels, and of bonding material, when bonding material is used, shall be supplied with at least ten equally spaced temperature measuring devices for determining the mean surface temperature.

23.2 Calibration.

23.21 Calibration of the calorimeter shall be performed by supplying enough heat to it to raise the temperature of the evaporator surface about 25°F. above the ambient. When internally located heaters are used, the evaporator shall be filled with refrigerating machine oil to serve as a heat transfer medium. In this case the oil shall be thoroughly cleaned out after the calibration. The ambient temperature shall be held constant within ± 2°F. at any desired value not exceeding 90°F. The heat input to the calorimeter shall be maintained constant, and temperature readings shall be taken at one hour intervals and continued until four successive averages do not differ by more than 1°F.

23.22 The heat leakage coefficient is given by:

$$q_l = \frac{Q_h}{t_e - t_a}$$

q_l = calorimeter heat leakage coefficient, in Btu. per hr. per °F. difference.

Q_h = calorimeter heat input, in Btu. per hr.

t_a = Average ambient temperature around the calorimeter in °F.

t_s = average surface temperature of tubes, and heaters if externally located, in °F.

23.23 The heat leakage is given by:

$$Q_l = q_l\ (t_a - t_s)$$

Q_l = heat leakage into calorimeter in Btu. per hr.

q_l = heat leakage coefficient in Btu. per hr. per °F. difference.

t_a = average ambient temperature around the calorimeter, in °F.

t_s = average surface temperature of tubes, and heaters, if externally located, in °F.

23.3 Procedure.

23.31 The specified suction pressure at the condensing unit is adjusted by means of the refrigerant control, and the specified temperature of the suction vapor at the condensing unit is adjusted by varying the heat input to the calorimeter.

23.32 Calorimeter data to be recorded after the condensing unit rating specified conditions have become established are:

a. Pressure of refrigerant vapor at evaporator outlet.

b. Temperature of refrigerant vapor at evaporator outlet.

c. Pressure of liquid refrigerant entering refrigerant control.

d. Temperature of liquid refrigerant entering refrigerant control.

e. Ambient temperature at calorimeter.

f. Calorimeter heater input.

g. Mean surface temperature of tubes, and heaters if externally located.

23.33 All readings shall be taken every 20 min.

23.34 Test shall be continued until four successive readings are within the allowable limits given in Section 16 and the variation in heat input to the calorimeter during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

23.4 Calculations.

23.41 The capacity of the condensing unit shall be determined from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\ (Q_h + Q_l)$$

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit in Btu. per lb.

h_{f2} = heat content of the refrigerant liquid entering the refrigerant control in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units in Btu. per lb.

h_{g2} = heat content of the refrigerant leaving the calorimeter in Btu. per lb.

Q = condensing unit capacity in Btu. per hr.

Q_h = calorimeter heater input in Btu. per hr.

Q_l = heat leakage into calorimeter in Btu. per hr.

24. *Liquid Refrigerant Quantity Method*

24.1 Description.

24.11 The apparatus for measuring liquid refrigerant volume shall consist of two vertical pressure vessels (about 4 ft. high) connected in parallel by horizontal headers at the top and bottom and shall have shut-off valves between the headers and vessels on the lines entering and leaving each vessel. The vessels and their connections from the condensing unit shall be thermally insulated. The pressure is equalized at all times by means of a small vapor line connecting the tops of the two vessels. Recommended sizes of vessels are given in Table below:

Recommended Sizes of Refrigerant Vessels

Diameter, in.	*Length, ft.*	*Range of use, tons*	*Refrigerant*
4	4	0 to 2	Freon (F 12)
4	4	0 " 2	Sulfur dioxide
4	4	0 " 2	Methyl chloride
4	4	2 " 7½	Ammonia
8	4	2 " 10	Freon (F 12)
8	4	2 " 10	Sulfur dioxide
8	4	2 " 10	Methyl chloride
8	4	7½ " 40	Ammonia
12	4	10 " 25	Freon (F 12)
12	4	10 " 25	Sulfur dioxide
12	4	10 " 25	Methyl chloride

24.12 The liquid refrigerant line leaving each vessel shall be equipped with a temperature measuring instrument well.

24.13 Each vessel shall be fitted with a graduated gage glass covering approximately the entire length.

24.14 A valve shall be located in the liquid refrigerant line from the condensing unit receiver to the calibrating vessels to regulate the flow of liquid and maintain the normal operating level in the receiver.

24.2 Calibration.

24.21 Each measuring vessel shall be calibrated with a liquid of known density and the gage glass scales marked in accordance with the calibration.

24.22 Calibrations are recommended for various sizes of vessels, as follows:

4-in. diameter, in hundredths (1/100) cu. ft.
8-in. diameter, in tenths (1/10) cu. ft.
12-in. diameter, in fourths (¼) cu. ft.

24.3 Procedure.

24.31 Measurement of the flow of liquid refrigerant is made by timing the interval for the collection of a volume of liquid for a period of not less than two minutes. The diameter of the vessel used should be such that the height of liquid refrigerant accumulated is not less than 6 in.

This volume is collected from the condensing unit by closing the outlet valve on the collecting vessel and the inlet valve on the other vessel which then continues the supply of refrigerant to the refrigerant control. Care must be taken to start timing after a steady flow is established and to finish timing before the level in the vessel supplying the refrigerant control falls to a point which breaks the liquid seal at the outlet. Timing shall be recorded on the vessel in which the liquid level is rising. The total volume of liquid refrigerant in the two vessels must be constant within 1 per cent at all times during the test. This total volume shall be determined by a simultaneous reading of the liquid levels in the vessels.

24.32 The data to be recorded after unit rating specified conditions have become established are:

a. Time for collection of liquid refrigerant volume.

b. Temperature of liquid refrigerant in collecting vessel.

c. Volume of liquid refrigerant collected.

24.33 All readings should be taken every 20 min.

24.34 Test shall be continued until four successive readings are within the allowable limits given in Section 16 and the variation in rate of liquid refrigerant flow during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

24.35 A sample of the liquid refrigerant shall be taken to determine the proportion of oil present.

24.4 Calculations.

24.41 The capacity of the condensing unit shall be determined from the following:

$$Q = Vd\,[(1 - 0.018\,wd)\,(h_{g1} - h_{f1}) - 0.44\,w\,(t_{f1} - t_{g1})]$$

d = density of liquid refrigerant, for the temperature at which flow is measured, in lb. per cu. ft.

h_{f1} = heat content of the refrigerant liquid leaving the condensing unit, in Btu. per lb.

h_{g1} = heat content of the refrigerant vapor entering the condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

Q = condensing unit capacity, in Btu. per hr.

t_{f1} = temperature of the refrigerant liquid leaving the condensing unit, in °F.

t_{g1} = temperature of the refrigerant vapor entering the condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in °F.

V = flow of mixture of the liquid refrigerant and oil in cu. ft. per hr.

w = weight of oil per unit weight of mixture of the liquid refrigerant and oil, in lb. per lb.

25. *Liquid Refrigerant Quantity Meter and Flow Meter Methods*

25.1 Description.

25.11 The liquid refrigerant meter may be either a quantity meter which consists of a metering mechanism accurate to within 1 per cent for recording refrigerant flow on a suitable counter graduated in gallons or cubic feet, or a flow meter which consists of a mechanism with suitable supplementary devices for determining the instantaneous rate of flow of the liquid refrigerant in cubic feet or gallons per hour.

25.12 The liquid refrigerant quantity meter shall be connected in the liquid line between the liquid receiver outlet and the refrigerant control.

25.13 In order that the meter may function properly under all conditions and that it may be protected from abuse and low refrigerant charge, additional apparatus shall be provided, preferably as an integral part of the quantity meter assembly, as follows:

a. Subcooling coil ahead of the meter sufficient to prevent vaporization of refrigerant through the meter.

b. Sight glasses located immediately before the subcooling coil and just beyond the meter to determine that vapor bubbles are not mixed with the liquid refrigerant.

c. Bypass valve and tube for bypassing the liquid around the meter. This valve shall be open except when data are being recorded.

d. Thermometer and well for measuring the liquid refrigerant temperature prior to its entering the subcooling coil.

e. Thermometer and well for measuring the temperature at which the liquid refrigerant enters the meter.

f. Pressure gage connected to the outlet side of the meter.

25.14 The calorimeter and its supply of heat load should be such as to balance out quickly and require little adjustment.

25.15 A supply of cold water at reasonably constant pressure and temperature together with a regulating valve should be provided for the meter subcooling coil.

25.2 Calibration.

25.21 The meter shall be calibrated periodically.

25.22 The testing fluid used in calibrating the meter shall have an absolute viscosity not less than one-half nor more than twice that of the liquid refrigerant whose quantity it normally measures.

25.23 The meter shall be calibrated at not less than three rates of flow covering the capacity range used.

25.3 Procedure.

25.31 With the meter bypass valve open, start the condensing unit. After the condensing unit rating specified conditions have become established, prepare for the test as follows:

a. Close the meter bypass valve.

b. Keep the liquid refrigerant subcooled at least

5°F. below its saturation temperature at the outlet from the flow meter.

25.32 After the above specified conditions have been obtained, record data on the meter and cooling units as follows:

a. Meter reading.

b. Saturation temperature of the refrigerant determined from the gage connected to the outlet side of the meter.

c. Temperature of the liquid entering the subcooling coil.

d. Temperature of the liquid entering the meter.

25.33 All readings shall be taken every 10 min.

25.34 Tests shall be continued until seven successive readings are within the allowable limits given in Section 16 and the variation in rate of liquid refrigerant flow during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

25.35 A sample of the liquid refrigerant shall be taken to determine the proportion of oil present.

25.4 Calculations.

25.41 The condensing unit capacity shall be determined from the following:

$$Q = Vd\,[(1 - 0.018\,wd)\,(h_{g1} - h_{f1}) - 0.44\,w\,(t_{f1} - t_{g1})]$$

$d =$ density of liquid refrigerant, for the temperature at which flow is measured, in lb. per cu. ft.

$h_{f1} =$ heat content of the refrigerant liquid leaving the condensing unit, in Btu. per lb.

$h_{g1} =$ heat content of the refrigerant vapor entering the condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

$Q =$ condensing unit capacity, in Btu. per hr.

$t_{f1} =$ temperature of the refrigerant liquid leaving the condensing unit, in °F.

$t_{g1} =$ temperature of the refrigerant vapor entering the condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in °F.

$V =$ flow of mixture of the liquid refrigerant and oil in cu. ft. per hr.

$w =$ weight of oil per unit weight of mixture of the liquid refrigerant and oil, in lb. per lb.

26. *Condenser Water Method*

26.1 Description.

26.11 The condenser is that which forms a part of the water-cooled condensing unit being tested. The calorimeter completes the equipment needed except for the provisions that must be made for taking the readings listed in paragraph 26.31.

26.12 This method shall be used *only* when the calculated heat leakage from the condenser to the 90°F. ambient is *less than 2 per cent* of the refrigerating effect of the condensing unit.

26.2 Calibration.

None.

26.3 Procedure.

26.31 During the primary test, record shall be made of:

a. Pressure of vapor refrigerant entering condenser.

b. Temperature of vapor refrigerant entering condenser.

c. Pressure of liquid refrigerant leaving condenser.

d. Temperature of liquid refrigerant leaving condenser.

e. Temperature of water entering condenser.

f. Temperature of water leaving condenser.

g. Ambient temperature surrounding condenser.

h. Quantity of condenser cooling water.

i. Average temperature of external condenser surface exposed to ambient.

26.32 All readings shall be taken concurrently and at the same time intervals specified in the primary test, and shall be continued until four successive readings are within the allowable limits given in Sections 14 and 16. The variation in condensing water temperature rise and rate of flow during the test shall not be great enough to cause an error of more than 1 per cent in condensing unit capacity.

26.4 Calculations.

26.41 The capacity of the condensing unit shall be calculated as follows:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g3} - h_{f3}}\,[W\,(t_2 - t_1) + Q_n]$$

$h_{f1} =$ heat content of refrigerant liquid leaving condensing unit, in Btu. per lb.

$h_{f3} =$ heat content of refrigerant liquid leaving the condenser, in Btu. per lb.

$h_{g1} =$ heat content of refrigerant vapor entering condensing unit, under the conditions specified in the A. S. R. E. Standard Method of Rating Mechanical Condensing Units, in Btu. per lb.

$h_{g3} =$ heat content of refrigerant vapor entering condenser, in Btu. per lb.

$Q =$ condensing unit capacity, in Btu. per hr.

$Q_n =$ heat loss from condenser to surrounding air, in Btu. per hr., $S\,U_a\,(t_e - t_a)$ approximately.

$S =$ outside surface of condenser, in sq. ft.

$t_a =$ ambient temperature, in °F.

$t_e =$ external surface temperature of condenser, in °F.

$t_1 =$ condenser entering water temperature, in °F.

$t_2 =$ condenser leaving water temperature, in °F.

$U_a =$ air film heat transfer coefficient, in Btu. per hr. per sq. ft. per °F.

$W =$ flow of condenser cooling water, in lb. per hr.

ASA
Reg. U. S. Pat. Off.

ASRE Standard 15-R
ASA B9.1-1950
UDC 621.56:614.8

American Standard

SAFETY CODE FOR MECHANICAL REFRIGERATION

SPONSOR—American Society of Refrigerating Engineers

Approved by ASRE Council, March, 1950

Approved by American Standards Association, Incorporated, May 24, 1950

Approved by the American Society of Refrigerating Engineers,
June 5, 1950

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40th STREET NEW YORK 18, N. Y.

Price $1.00

American Standards Association Sectional Committee B9

Organization	*Names*	
American Society of Refrigerating Engineers	George B. Bright	
	P. B. Christensen	
	Austin S. Ford	
	W. R. Hainsworth	
	Glenn Muffly	
	S. R. Hirsch	Alternate
Air Conditioning & Refrigerating Machinery Association	D. M. Abel	
	E. T. Benson	
	Frank H. Faust	
	L. S. Morse	
	Leonard C. Bastian	Alternate
	J. R. Chamberlain	Alternate
	William B. Henderson	Alternate
	W. W. Higham	Alternate
	P. R. Lynn	Alternate
	A. B. Newton	Alternate
Air Conditioning and Refrigeration Contractors Association	H. J. Prebensen	
	Theodore A. Reina	Alternate
American Gas Association	W. S. Walker	
American Institute of Chemical Engineers	Crosby Field	
American Institute of Electrical Engineers	W. C. Goodwin	
American Institute of Refrigeration	Fred Ophuls	
American Meat Institute	John S. Bartley	
	Henry D. Tefft	
American Society of Heating & Ventilating Engineers	Lee Nusbaum	
	Dexter J. Purinton	Alternate
American Society of Mechanical Engineers	D. L. Fiske	
	E. W. Gallenkamp	
	Carl F. Kayan	
	A. W. Oakley	Alternate
American Society of Safety Engineers	C. C. Ruddick	
Association of Casualty & Surety Companies, Accident Prevention Dept.	Robert Stickler	
Building Officials Conference of America Inc.	J. C. Rehard	
Bureau of Explosives	R. S. Hunt	
Compressed Gas Association Inc.	Franklin R. Fetherston	
Electric Light & Power Group	S. S. Sanford	
	C. K. Poarch	Alternate
Frozen Food Lockers Institute	L. O. Warner	
Heating Piping & Air Conditioning Contractors National Assn.	Alfred C. Buensod	
	A. F. Nass	Alternate
International Acetylene Association	H. D. Edwards	
	M. G. Farrar	Alternate
International Association of Government Labor Officials	Robert M. Murray	
	Michael P. Riccitelli	Alternate
Manufacturers Standardization Society of the Valve & Fittings Industry	W. P. Kliment	
	L. W. Kattelle	Alternate
Manufacturing Chemists' Association of the U. S.	H. L. Miner	
	R. J. Thompson	Alternate
National Association of Ice Industries	W. A. Schmid	
National Association of Mutual Casualty Companies	Representation Vacant	
	J. C. Stennett	Alternate
National Association of Refrigerated Warehouses, a Div. of American Warehousemen's Assn.	Herbert T. Walworth	Alternate
	P. B. Christensen	
	J. C. Irwin	Alternate
National Bureau of Standards, U. S. Department of Commerce	John A. Dickinson	
	R. S. Dill	Alternate
National Electrical Manufacturers Association	Byron S. Booher	
	P. H. Lawrenz	
	Milton Kalischer	Alternate
	S. H. Mortensen	Alternate
	F. L. Tarleton	Alternate
	W. M. Timmerman	Alternate
National Fire Protection Association	H. E. Newell	
	E. W. Fowler	Alternate
National Retail Dry Goods Association	D. F. Hahn	
National Safety Council	H. D. Edwards	
	Emerson Brandt	Alternate
New York Board of Fire Underwriters	William G. Hayne	
New York City Bureau of Fire Prevention	George Sklenarik	
Refrigeration Equipment Manufacturers Association	C. M. Brown	
	Lee C. Love	
	G. E. Graff	Alternate
	Harry Pendergast	Alternate
Refrigeration Service Engineers Society	L. J. Wallace	
Underwriters' Laboratories Inc.	S. V. James	
	C. W. Rulon	Alternate
U. S. Department of Labor	Henry E. Kaltun	

American Standards Association Sectional Committee B9

WILLIAM R. HAINSWORTH, *Chairman*
H. D. EDWARDS, *Vice Chairman*
E. T. BENSON, *Vice Chairman*
HENRY G. LAMB, *Secretary*

American Society of Refrigerating Engineers' Representatives:

William R. Hainsworth
George B. Bright
P. B. Christensen
Austin S. Ford
Glenn Muffly
S. R. Hirsch (Alternate)

Subcommittee Personnel

Code Drafting Subcommittee

Glenn Muffly, *Chairman*
B. S. Booher
J. R. Chamberlain
W. P. Kliment
F. H. Faust, *Vice Chairman*
A. C. Buensod
E. W. Fowler
L. J. Wallace, succeeding C. C. E. Harris

Special Committee on Specifications Related to ASME Unfired Pressure Vessels Code

J. R. Chamberlain, ***Chairman***
F. H. Faust
Glenn Muffly
E. T. Benson
W. P. Kliment

Safety Valves Subcommittee

J. R. Chamberlain, *Chairman*
C. S. Cragoe
W. P. Kliment
S. V. James
F. R. Fetherston
C. O. Myers

Refrigerant Piping Valves & Fittings Subcommittee

W. P. Kliment, *Chairman*
Lars Hanson
L. J. Wallace
A. C. Buensod
P. B. Reed
D. K. McIlvaine

Unfired Pressure Vessels Subcommittee

F. R. Fetherston, *Chairman*
W. A. Grant
L. S. Morse
Lee Nusbaum
J. F. Scott

Relations Subcommittee

E. T. Benson, *Chairman*
H. F. Spoehrer
A. C. Buensod
L. J. Wallace

The above subcommittees were responsible for the revision of the ASA B9-1939 Safety Code which resulted in this ASA B9.1-1950 Safety Code.

These committees have been discharged and the following Interpretations Subcommittee now has the responsibility of interpreting the provisions of this code and for submitting recommendations to the Sectional Committee for future revisions of the code.

Interpretations Subcommittee

E. T. Benson, *Chairman*
J. R. Chamberlain
S. V. James
J. C. Rehard
L. J. Wallace
M. C. Turpin, Secretary

CONTENTS

Section 1 • Scope and Purpose
- 1.1 Scope 5
- 1.2 Purpose 5
- 1.3 Application 5

Section 2 • Definitions 5

Section 3 • Building Occupancy Classification
- 3.1 Locations 7
- 3.2 Institutional Occupancy 7
- 3.3 Public Assembly Occupancy 7
- 3.4 Residential Occupancy 8
- 3.5 Commercial Occupancy 8
- 3.6 Industrial Occupancy 8
- 3.7 Mixed Occupancy 8

Section 4 • Refrigeration System Classification by Type
- 4.1 Refrigerating Systems 8
- 4.2 Direct System 8
- 4.3 Indirect System 8
- 4.4 Double (or Secondary) Refrigerant System 8

Section 5 • Refrigerant Classification 8

Section 6 • Requirements for Institutional, Public Assembly, Residential, and Commercial Occupancies
- 6.1 General 9
- 6.2 Group 1 Refrigerants 9
- 6.3 Group 2 Refrigerants 10
- 6.4 Group 3 Refrigerants 11

Section 7 • Requirements for Industrial Occupancies
- 7.1 General 11
- 7.2 Number of Persons 11

Section 8 • Installation Requirements
- 8.1 Foundations and Supports 11
- 8.2 Moving Machinery 11
- 8.3 Clear Space 11
- 8.4 Condensing Units or Compressor Units with Enclosures 11
- 8.5 Water Supply and Discharge Connections 11
- 8.6 Illumination 11
- 8.7 Electrical Equipment and Wiring 11
- 8.8 Gas Fuel Devices and Equipment 11
- 8.9 Open Flames 11
- 8.10 Flammable Refrigerants 11
- 8.11 Machinery Room Requirements 11
- 8.12 Air Duct Systems 12

Section 9 • Refrigerant Piping, Valves, Fittings, and Related Parts
- 9.1 General 12
- 9.2 Metal Enclosures or Pipe Ducts for Soft Copper Tubing 12
- 9.3 Specific Minimum Requirements for Refrigerant Pipe and Tubing 12
- 9.4 Joints and Refrigerant-Containing Parts in Air Ducts 12
- 9.5 Exposure of Refrigerant Pipe Joints 12
- 9.6 Stop Valves 13
- 9.7 Location of Refrigerant Piping 13

Section 10 • Design and Construction of Equipment
- 10.1 General 13
- 10.2 Minimum Test Pressures 13
- 10.3 Equipment Listed by an Approved Nationally Recognized Testing Laboratory 13
- 10.4 Pressure-Limiting Devices 13
- 10.5 Liquid Level Gage Glasses 13
- 10.6 Dial of a Pressure Gage 14
- 10.7 Nameplate 14

Section 11 • Refrigerant-Containing Pressure Vessels
- 11.1 Refrigerant-Containing Pressure Vessels Exceeding 6 Inches in Diameter 14
- 11.2 Refrigerant-Containing Pressure Vessels Not Exceeding an Inside Diameter of 6 Inches 14
- 11.3 Safety Devices 14
- 11.4 Standard Hydrostatic Tests 14

Section 12 • Relief Devices in General
- 12.1 General 14
- 12.2 Pressure-Relief Devices for Positive Displacement Compressors 15
- 12.3 Discharge of Pressure-Relief Devices and Fusible Plugs 15
- 12.4 Ammonia Discharge 15
- 12.5 Sulphur Dioxide Discharge 15

Section 13 • Relief Devices for Pressure Vessels
- 13.1 General 16
- 13.2 Pressure Vessels over 5 Cubic Feet 16
- 13.3 Pressure Vessels with Gross Volume of 5 Cubic Feet or Less 16
- 13.4 Relief Devices for Pressure Vessels Used as, or as Part of, Evaporators 16
- 13.5 Required Capacity 16
- 13.6 Pressure-Relief Device Setting 16
- 13.7 Rupture Member Setting When Used in Parallel with Relief Valves 16
- 13.8 Other Rupture Member Setting 16
- 13.9 Marking of Relief Devices 16

Section 14 • Field Tests
- 14.1 General 16
- 14.2 Test Medium 16
- 14.3 Posting of Tests 16

Section 15 • Instructions
- 15.1 Signs 17
- 15.2 Metal Signs for Systems Containing More Than 100 Pounds of Refrigerant 17
- 15.3 New Sign for Changed Refrigerant 17
- 15.4 Charging and Discharging Refrigerants .. 17
- 15.5 Refrigerants Withdrawn from Refrigerating Systems 17
- 15.6 Containers Used for Refrigerants Withdrawn from a Refrigerating System 17
- 15.7 Substitution of Kind of Refrigerant 17
- 15.8 Refrigerant Stored in a Machinery Room 18
- 15.9 Masks or Helmets 18
- 15.10 Maintenance 18
- 15.11 Responsibility as to Operation of the System 18
- 15.12 Pressure Gages 18

Appendix 19

LIST OF TABLES

1. Maximum Permissible Quantities of Group 1 Refrigerants for Direct Systems 9
2. Maximum Permissible Quantities of Group 2 Refrigerants for Direct Systems 10
3. Maximum Permissible Quantities of Group 2 Refrigerants for Indirect Systems 10
4. Maximum Permissible Quantities of Flammable Refrigerants 11
5. Minimum Air Duct Areas and Openings 12
6. Minimum Refrigerant Leak Field Test Pressures 17
7. Length of Discharge Piping for Relief Valves or Rupture Members of Various Discharge Capacities 18

American Standard

Safety Code for Mechanical Refrigeration

FOREWORD

This Safety Code for Mechanical Refrigeration is sponsored by the American Society of Refrigerating Engineers under the rules and regulations of the American Standards Association Inc. as Project B9.

Complete administrative details are not provided as they will necessarily vary in different states and municipalities.

Most of the refrigeration-producing equipment, household, commercial, or industrial, is manufactured by companies whose business is nationwide. It is, therefore, desirable that this Code be universally adopted so that safety provisions for refrigerating installations will be standardized, thereby permitting quantity production of refrigerating equipment with minimum cost to the user.

A standing Sectional Committee is provided to keep the provisions of this Code abreast of the advancements in the art of refrigeration.

Words *italicized* in the text have been specifically defined in Section 2 and are used in accordance with those definitions.

Section 1. SCOPE AND PURPOSE

cope. The application of this Code is intended
sure the safe design, construction, installation,
ation, and inspection of every *refrigerating*
em employing a fluid which is vaporized and
rmally liquefied in its refrigerating cycle, when
oyed under the occupancy classifications listed
ection 3. The provisions of this Code are not
ded to apply to the use of water or air as a
gerant.

urpose. This Code is intended to provide rea-
ble safeguards to life, limb, health, and prop-
; to correct certain practices which are incon-
nt with safety; and to prescribe standards of
y which will properly influence future progress
developments in *refrigerating systems.* It is
intent of this Code to recognize listing by *an*
oved nationally recognized testing laboratory,
efined in 2.3.1, as an alternate method of pro-
g equivalent safety in certain instances (10.3-
13.3).

pplication. This Code shall apply to *refrigerat-*
systems installed subsequent to its adoption
to parts replaced or added to systems installed
or subsequent to its adoption. In cases of
tical difficulty or unnecessary hardship, the
ority having jurisdiction may grant exceptions
the literal requirements of this Code or per-
the use of other devices or methods, but only
it is clearly evident that equivalent protection
ereby secured.

TE: To secure the uniform application of this Code,
rities having jurisdiction are urged, before rendering
ons on disputed points, to consult the committee which
ulated it—the Committee on Safety Code for Mechani-
efrigeration, B9, in care of the American Standards
iation, Incorporated, 70 East 45th Street, New York
17, N. Y., or the American Society of Refrigerating Engineers, 40 West 40th Street, New York 18, N. Y.

Section 2. DEFINITIONS

2.1 *Absorber (Adsorber)* is that part of the *low side* of an *absorption system* used for absorbing (adsorbing) vapor *refrigerant.*

2.2 *Absorption System*—see 2.51.1.

2.3 *Approved* means acceptable to the authorities having jurisdiction.

2.3.1 *An Approved Nationally Recognized Testing Laboratory* is one acceptable to the authorities having jurisdiction, that provides uniform testing and examination procedures under established standards, is properly organized, equipped and qualified for testing, and has a follow-up inspection service of the current production of the listed products.

2.4 *Brazed Joint,* for the purpose of this code, is a gas-tight joint obtained by the joining of metal parts with alloys which melt at temperatures higher than 1000 F but less than the melting temperatures of the joined parts.

2.5 *Brine* is any liquid, used for the transmission of heat without a change in its state, having no flash point or a flash point above 150 F determined by American Society for Testing Materials method D93-46. (See Appendix 1.)

2.6 *Centrifugal Compressor* is a *nonpositive displacement compressor* which depends, at least in part, on centrifugal force for pressure rise. (See 2.40.)

2.7 *Centrifugal System*—see 2.51.2.

2.8 *Compressor* is a specific machine, with or without accessories, for compressing a given *refrigerant* vapor.

2.9 *Compressor Unit* is a *condensing unit* less the *condenser and liquid receiver.*

2.10 *Condenser* is a vessel or arrangement of pipe or tubing in which vaporized *refrigerant* is liquefied by the removal of heat.

2.11 *Condensing Unit* is a specific refrigerating machine combination for a given *refrigerant*, consisting of one or more power-driven *compressors*, *condensers, liquid receivers* (when required), and the regularly furnished accessories.

2.12 *Container* is a cylinder for the transportation of *refrigerant.* (See Appendix 2.)

2.13 *Department Store* is the entire space occupied by one *tenant* or more than one *tenant* in an individual store where more than 100 persons commonly assemble on other than the street-level floor for the purpose of buying personal wearables and other merchandise.

2.14 *Design Working Pressure* is the maximum allowable working pressure for which a specific part of a system is designed.

2.15 *Direct System*—see 4.2.

2.16 *Double Indirect Vented Open-Spray System*—see 4.3.4.

2.17 *Double (or Secondary) Refrigerant System*—see 4.4.

2.18 *Duct* is a tube or conduit used for conveying or encasing purposes as specifically defined below:

a) *Air duct* is a tube or conduit used for conveying air. (The air passages of *self-contained systems* are not to be construed as *air ducts.)*
b) *Pipe duct* is a tube or conduit used for encasing pipe.
c) *Wire duct* is a tube or conduit used for encasing either moving or stationary wire, rope, etc.

2.19 *Entrance* is a confined passageway immediately adjacent to the door through which people enter a building.

2.20 *Evaporator* is that part of the system in which liquid *refrigerant* is vaporized to produce refrigeration.

2.21 *Exit* is a confined passageway immediately adjacent to the door through which people leave a building.

2.22 *Expansion Coil* is an *evaporator* constructed of pipe or tubing.

2.23 *Fusible Plug* is a device having a predetermined-temperature fusible member for the relief of pressure.

2.24 *Generator* is any device equipped with a heating element used in the *refrigerating system* to increase the pressure of *refrigerant* in its gas or vapor state for the purpose of liquefying the *refrigerant.*

2.25 *Hallway* is a corridor for the passage of people.

2.26 *High Side* means the parts of a *refrigerating system* under *condenser* pressure.

2.27 *Humanly Occupied Space* is a space normally frequented or occupied
machinery rooms and walk-in coolers used
marily for refrigerated storage.

2.28 *Indirect Closed-Surface System*—see 4.3.2

2.29 *Indirect Open-Spray System*—see 4.3.1.

2.30 *Indirect System*—see 4.3.

2.31 *Indirect Vented Closed-Surface System*—
4.3.3.

2.32 *Liquid Receiver* is a vessel permanently
nected to a system by inlet and outlet pipes
storage of a liquid refrigerant.

2.33 *Lobby* is a waiting room, or large *hall*
serving as a waiting room.

2.34 *Low Side* means the parts of a *refrigera*
system under *evaporator* pressure

2.35 *Machinery* is the refrigerating equipm
forming a part of the *refrigerating system* inc
ing any or all of the following: *compressor,*
denser, generator, absorber (adsorber), liquid
ceiver, connecting pipe, or *evaporator.*

2.36 *Machinery Room* is a room in which a
frigerating system is permanently installed
operated but not including *evaporators* locate
a cold storage room, refrigerator box, air co
space, or other enclosed space. Closets solely
tained within, and opening only into, a room
not be considered *machinery rooms* but *shal*
considered a part of the *machinery room* in w
they are contained or open into. It is not the
tent of this definition to cause the space in w
a *self-contained system* is located to be classifie
a *machinery room.* (See 8.11.)

2.37 *Machinery Room, Class T* is a room ha
machinery but no flame-producing appar
permanently installed and operated and also
forming to the following:

a) Any doors, communicating with the build
shall be *approved* self-closing, tight-fit
fire doors.
b) Walls, floor, and ceiling *shall* be tight an
not less than one-hour fire-resistive const
tion.
c) It *shall* have an *exit* door which opens
rectly to the outer air or through a vestil
type *exit* equipped with self-closing, t
fitting doors.
d) Exterior openings, if present, *shall no*
under any fire escape or any open stair
e) All pipes piercing the interior walls, ce
or floor of such room *shall* be tightly s
to the walls, ceiling, or floor through w
they pass.
f) Emergency remote controls to stop the a
of the *refrigerant compressor shall* be
vided and located immediately outside
machinery room.
g) Mechanical means *shall* be provided for
tilation. (See 8.11.3.)
h) Emergency remote controls for the mec
cal means of ventilation *shall* be pro
and located outside the *machinery roo*

2.38 *Manufacturer* is, for the purpose of this
the company or organization which evidenc

sibility by affixing its name or nationally
ered trade-mark or trade name to the re-
ation equipment concerned.

echanical Joint, for the purpose of this code,
gas-tight joint, obtained by the joining of
parts through a positive-holding mechanical
uction.

onpositive Displacement Compressor is a
ressor in which increase in vapor pressure is
ed without changing the internal volume of
mpression chamber.

iping means the pipe or tube mains for inter-
cting the various parts of a *refrigerating*
n.

ositive Displacement Compressor is a *com-*
or in which increase in vapor pressure is at-
by changing the internal volume of the
ression chamber.

ressure-Imposing Element is any device or
n of the equipment used for the purpose of
sing the *refrigerant* vapor pressure.

ressure-Limiting Device is a pressure-re-
ive mechanism designed to automatically stop
peration of the *pressure-imposing element* at
determined pressure.

ressure-Relief Device is a pressure-actuated
or *rupture member* designed to automatically
e excessive pressure.

ressure-Relief Valve is a pressure-actuated
held closed by a spring or other means and
ned to automatically relieve pressure in ex-
f its setting.

ressure Vessel is any refrigerant-containing
tacle of a *refrigerating system,* other than
rators, each separate section of which does
xceed ½ cubic foot of refrigerant-containing
e, *expansion coils, compressors,* controls,
rs, pipe, and pipe fittings.

eceiver—see 2.32.

eciprocating Compressor is a *positive dis-*
ment compressor with a piston or pistons
g in a straight line but alternately in opposite
ions.

efrigerant is a substance used to produce
eration by its expansion or vaporization.

efrigerating System is a combination of in-
nected refrigerant-containing parts consti-
one closed *refrigerant* circuit in which a
erant is circulated for the purpose of extract-
eat. (See Section 4 for classification of *re-*
ating systems by type.)

.1 *Absorption System* is a *refrigerating sys-*
which the gas evolved in the *evaporator* is
up by an *absorber* or *adsorber.*

.2 *Centrifugal System* is a specific combina-
f *machinery* for a given *refrigerant,* consist-
a power-driven *centrifugal compressor,* a
nser, a water or other liquid cooler, and the
rly furnished accessories. The component
after assembly and testing at the factory,
ually disassembled for shipment.

.3 *Sealed Absorption System* is a *unit system*
for Group 2 *refrigerants* only in which all refrigerant-containing parts are made permanently tight by welding or brazing against *refrigerant* loss. (This is a restrictive definition for the purposes of this code as used in 6.1.2 and 6.3.1.) (See 2.51.1.)

2.51.4 *Self-Contained System* is a complete factory-made and factory tested system in a suitable frame or enclosure which is fabricated and shipped in one or more sections and in which no refrigerant-containing parts are connected in the field other than by companion or block valves.

2.51.5 *Unit System* is a *self-contained system* which has been assembled and tested prior to its installation and which is installed without connecting any refrigerant-containing parts. A *unit system* may include factory-assembled companion or block valves.

2.52 *Rupture Member* is a device that will automatically rupture at a predetermined pressure.

2.53 *Shall.* Where *"shall"* or *"shall not"* is used for a provision specified, that provision is intended to be mandatory.

2.54 *Should. "Should"* or *"it is recommended"* is used to indicate provisions which are not mandatory but which are pointed out here as recommended good practice.

2.55 *Sealed Absorption System*—see 2.51.3.

2.56 *Self-Contained System*—see 2.51.4.

2.57 *Soldered Joint,* for the purpose of this code, is a gas-tight joint obtained by the joining of metal parts with metallic mixtures or alloys which melt at temperatures below 1000 F and above 400 F.

2.58 *Stop Valve* is a shut-off for controlling the flow of *refrigerant.*

2.59 *Tenant,* as herein used, shall be construed as a person, firm, or corporation possessed with the legal right to occupy premises.

2.60 *Unit System*—see 2.51.5.

2.61 *Welded Joint,* for the purpose of this code, is a gas-tight joint, obtained by the joining of metal parts in the plastic or molten state.

Section 3. BUILDING OCCUPANCY CLASSIFICATION

3.1 **Locations** in which *refrigerating systems* may be placed are grouped by occupancy as follows:

3.2 **Institutional Occupancy** *shall* apply to that portion of a building in which persons are confined to receive medical, charitable, educational, or other care or treatment, or in which persons are held or detained by reason of public or civic duty, including among others, hospitals, asylums, sanitariums, police stations, jails, court houses with cells, and similar occupancies.

3.3 **Public Assembly Occupancy** *shall* apply to that portion of the premises in which persons congregate for civic, political, educational, religious, social, or recreational purposes; including among others, armories, assembly rooms, auditoriums, ball rooms, bath houses, bus terminals, broadcasting studios, churches, colleges, court houses without cells, dance halls, *department stores,* exhibition halls, fraternity halls, libraries, lodge rooms, mortuary chapels, museums, passenger depots, schools, skating rinks, subway stations, theaters, and similar occupancies.

3.4 Residential Occupancy *shall* apply to that portion of a building in which sleeping accommodations are provided for more than two families, including among others, club houses, convents, dormitories, hotels, lodging houses, multiple storey apartments, studios, tenements, and similar occupancies.

3.5 Commercial Occupancy *shall* apply to that portion of a building used for the transaction of business; for the rendering of professional services; for the supplying of food, drink, or other bodily needs and comforts; for manufacturing purposes or for the performance of work or labor (except as included under 3.6 Industrial Occupancy) including among others, bake shops, fur storage, laboratories, loft buildings, markets, office buildings, professional buildings, restaurants, stores other than *department stores,* and similar occupancies.

3.6 Industrial Occupancy *shall* apply to an entire building when used by a single *tenant* for manufacturing, processing, or storage of materials or products, including among others, chemical, food, candy, and ice cream factories, ice making plants, meat packing plants, refineries, perishable food warehouses, and similar occupancies.

3.7 Mixed Occupancy *shall* apply to a building occupied or used for different purposes in different parts. When the occupancies are cut off from the rest of the building by tight partitions, floors, and ceilings and protected by self-closing doors, the requirements for each type of occupancy *shall* apply for its portion of the building. For example, the cold storage spaces in retail frozen food lockers, hotels, and *department stores* might be classified under Industrial Occupancy, whereas other portions of the buildings would be classified under other occupancies. When the occupancies are not so separated, the occupancy carrying the more stringent requirements *shall* govern.

Section 4. REFRIGERATING SYSTEM CLASSIFICATION BY TYPE

4.1 Refrigerating Systems (see 2.51) *shall* be divided into classes, descriptive of the method employed for extracting heat as follows in 4.2 to 4.4, inclusive. The *direct* and various *indirect systems* referred to are illustrated in Figure 1.

4.2 Direct System is one in which the *evaporator* is in direct contact with the material or space refrigerated or is located in air-circulating passages communicating with such spaces.

4.3 Indirect System is one in which a liquid, such as *brine* or water, cooled by the *refrigerant,* is circulated to the material or space refrigerated or is used to cool air so circulated. *Indirect systems* which are distinguished by the type or method of application are as given in the following paragraphs:

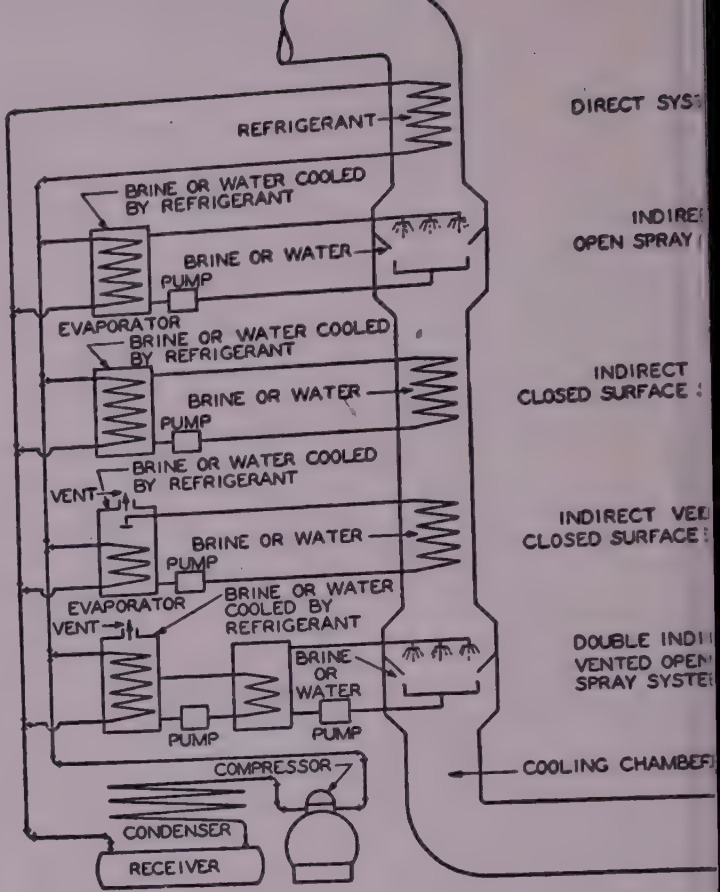


FIG. 1

4.3.1 *Indirect Open-Spray System* is one in wh liquid, such as *brine* or water, cooled by an e *rator* located in an enclosure external to a co chamber, is circulated to such cooling chambe is sprayed therein.

4.3.2 *Indirect Closed-Surface System* is one in a liquid, such as *brine* or water, cooled evaporator located in an enclosure external cooling chamber, is circulated to and through cooling chamber in pipes or other closed ci

4.3.3 *Indirect Vented Closed-Surface System* i in which a liquid, such as *brine* or water, coo an *evaporator* located in a vented enclosure ex to a cooling chamber, is circulated to and th such cooling chamber in pipes or other close cuits.

4.3.4 *Double Indirect Vented Open-Spray Sys* one in which a liquid, such as *brine* or water, by an *evaporator* located in a vented enclos circulated through a closed circuit to a seco closure where it cools another supply of a such as *brine* or water, and this liquid in t circulated to a cooling chamber and is sp therein.

4.4 Double (or Secondary) Refrigerant System is which an evaporative *refrigerant* is used secondary circuit. For the purpose of this each system enclosing a separate body of an rative *refrigerant shall* be considered as a se *direct system.*

Section 5. REFRIGERANT CLASSIFICATI

5.1 General. *Refrigerants* are, for the purp this code, divided into groups as follows:

.1.1 *Group 1*

Carbon dioxide	CO_2
Dichlorodifluoromethane (Freon-12)	CCl_2F_2
Dichloromethane (Carrene No. 1) (Methylene chloride)	CH_2Cl_2
Dichloromonofluoromethane (Freon-21)	$CHCl_2F$
Dichlorotetrafluoroethane (Freon-114)	$C_2Cl_2F_4$
Monochlorodifluoromethane (Freon-22)	$CHClF_2$
Trichloromonofluoromethane (Freon-11) (Carrene No. 2)	CCl_3F
Trichlorotrifluoroethane (Freon-113)	$C_2Cl_3F_3$

.1.2 *Group 2*

Ammonia	NH_3
Dichloroethylene	$C_2H_2Cl_2$
Ethyl chloride	C_2H_5Cl
Methyl chloride	CH_3Cl
Methyl formate	$HCOOCH_3$
Sulphur dioxide	SO_2

.1.3 *Group 3*

Butane	C_4H_{10}
Ethane	C_2H_6
Ethylene	C_2H_4
Isobutane	$(CH_3)_3CH$
Propane	C_3H_8

Section 6. REQUIREMENTS FOR INSTITUTIONAL, PUBLIC ASSEMBLY, RESIDENTIAL, AND COMMERCIAL OCCUPANCIES

.1 General

6.1.1 ***Public Stairway, Stair Landing, Entrance, or Exit.*** No *refrigerating system shall be installed* in or on a public stairway, stair landing, *entrance*, or *exit*.

6.1.2 ***Public Hallway or Lobby.*** No *refrigerating ystem shall* interfere with free passage. No Group 2 *refrigerant shall* be permitted in public *hallways* or *lobbies* of Institutional or Public Assembly Occupancies. *Refrigerating systems* installed in a public *hallway* or *lobby shall* be limited to:

a) *Unit Systems* containing not more than the quantities of a Group 1 *refrigerant* specified in Table 1, or

b) *Sealed Absorption Systems* containing not more than 3 pounds of Group 2 *refrigerant* when in Residential and Commercial Occupancies.

6.1.3 ***Refrigerant Piping Through Floors.*** *Refrigerant piping shall not* be carried through floors except as follows:

a) It may be carried from the basement to the first floor or from the top floor to a *machinery* penthouse or to the roof.

b) For the purpose of connecting to a *condenser* on the roof, it may be carried through an *approved*, rigid and tight continuous fire-resisting *pipe duct* or shaft having no openings on intermediate floors, or it may be carried on the outer wall of the building provided it is not located in an air shaft, closed court, or in other similar open spaces enclosed within the outer walls of the building.

c) In systems containing Group 1 *refrigerants*, the *refrigerant piping* may also be carried through floors, intermediate between the first floor and the top floor, provided it is enclosed in an *approved*, rigid, and tight continuous fire-resisting *pipe duct* or shaft where it passes through any intermediate space. Where the *refrigerating system* serves an air conditioning system, the *piping* need not be enclosed where it passes through air conditioned spaces served by that system. The *pipe duct* or shaft *shall* be vented to the outside or to a space served by the air conditioning system.

TABLE 1. Maximum Permissible Quantities of Group 1 Refrigerants for Direct Systems

Refrigerant name	*Chemical formula*	*Maximum quantity in lb per 1000 cu ft of humanly occupied space**
Carbon dioxide	CO_2	11
Dichlorodifluoromethane (Freon-12)	CCl_2F_2	31
Dichloromethane (Methylene chloride), (Carrene No. 1)	CH_2Cl_2	6
Dichloromonofluoromethane (Freon 21)	$CHCl_2F$	13
Dichlorotetrafluoroethane (Freon-114)	$C_2Cl_2F_4$	44
Monochlorodifluoromethane (Freon-22)	$CHClF_2$	22
Trichloromonofluoromethane (Freon-11)	CCl_3F	35
Trichlorotrifluoroethane (Freon-113)	$C_2Cl_3F_3$	24

*NOTES:

a) When the refrigerant-containing parts of a system are located in one or more enclosed spaces, the cubical content of the smallest enclosed *humanly occupied space* other than the *machinery room*, *shall* be used to determine the permissible quantity of *refrigerant* in the system.

b) When the *evaporator* is located in an *air duct* system, cubical content of the smallest humanly occupied enclosed space served by the *air duct* system *shall* be used to determine the permissible quantity of *refrigerant* in the system; however, if the air flow to any enclosed space served by the *air duct* system cannot be shut off or reduced below one-quarter of its maximum, the cubical contents of the entire space served by the *air duct* system may be used to determine the permissible quantity of *refrigerant* in the system.

c) Volatile charge in a control *shall not* be considered as *refrigerant*.

6.2 Group 1 Refrigerants

6.2.1 *Direct Systems.* The maximum permissible quantity of a Group 1 *refrigerant* in a *direct system* is specified in Table 1 except Institutional Occupancies where further limited by 6.2.1.1.

6.2.1.1 *Direct Systems* in Institutional Occupancies *shall* be limited to *unit systems* containing not more than 20 pounds of Group 1 *refrigerants*, except in kitchens, laboratories, and mortuaries. (See 6.2.4.)

6.2.1.2 In Institutional and Public Assembly Occupancies, direct *expansion coils* or *evaporators* used for air conditioning and located downstream from, and in proximity to, a heating coil, or located upstream within 18 inches of a heating coil, *shall* be fitted with a relief device discharging to the outside of the building in an *approved* manner; except that such a relief device *shall not* be required on *unit* or *self-contained systems* if the internal volume of the *low side* of the system which may be shut off by valves, divided by the total weight of *refrigerant* in the system less the weight of *refrigerant* vapor contained in the other parts of the system at 110 F, exceeds the specific volume of the *refrigerant* at critical conditions of temperature and pressure.

(NOTE: The above exemption is also stated in formula form below.)

$$\frac{V_1}{W_1 \;-\; W_2} \text{ \textit{shall} be more than } V_{sp}$$

where V_1 = *low side* volume, cu ft

V_{sp} = specific volume at critical conditions of temperature and pressure, cu ft per lb

W_1 = total weight of *refrigerant* in system, lb

W_2 = weight of *refrigerant* vapor (lb) at 110 F in V_2, or

$$= \frac{V_2}{\text{specific volume of \textit{refrigerant}, in cu ft per lb, at 110 F,}}$$

where V_2 = total volume of system less V_1, cu ft

6.2.2 *Indirect Systems.* A system containing more than the quantity of a Group 1 *refrigerant* allowed in Table 1 *shall* be of the *indirect* type with all refrigerant-containing parts, excepting parts mounted outside the building, installed in a *machinery room* used for no other purpose than for mechanical equipment.

6.2.3 *Open Flames in Machinery Rooms.* No open flame or apparatus to produce an open flame *shall* be installed in a *machinery room* where any *refrigerant* other than carbon dioxide is used unless the flame is enclosed and vented to the open air. The use of matches, cigarette lighters, halide leak detectors, and similar devices *shall* not be considered a violation of this paragraph or of 6.2.4.

6.2.4 *Open Flames in Institutional and Public Assembly Occupancies.* In Institutional and Public Assembly Occupancies, when more than 1 pound of a Group 1 *refrigerant,* other than carbon dioxide, is used in a system any portion of which is in a room where there is an apparatus for producing an open flame, then such *refrigerant shall* be classed in Group 2 unless the flame-producing apparatus is provided with a hood and flue capable of removing the products of combustion to the open air.

6.3 Group 2 Refrigerants

6.3.1 *Direct Systems.* The maximum permissible quantity of a Group 2 *refrigerant* in a *direct system* is shown in Table 2. (Also see 6.1.2.)

TABLE 2. Maximum Permissible Quantities of Group Refrigerants for Direct Systems*

Type of refrigerating system	*Maximum pounds for various occupancies**			
	Institutional	*Public assembly*	*Residential*	*Com-merci*
Sealed Absorption Systems:				
a) In public *hallways* or *lobbies*	0	0	3	3
b) In other than public *hallways* or *lobbies*	0**	6	6	20
Self-Contained or Unit Systems:				
a) In public *hallways* or *lobbies*	0	0	0	0
b) In other than public *hallways* or *lobbies*	0	0**	6	20
Other *Refrigerating Systems* in *Machinery Room* (see 2.36)	0	0	300	600
Other *Refrigerating Systems* in *Class T Machinery Room* (see 2.37)	500	1000	No limit	No lim

* *Direct systems* containing Group 2 *refrigerants shall n* be used for air conditioning for human comfort in Institutional, Public Assembly, and Residential Occupancies an *shall* be so used in Commercial Occupancies only when containing not more than 20 pounds of *refrigerant.*

**Six pounds allowed when installed in kitchens, laboratories, and mortuaries for applications other than air conditioning for human comfort.

6.3.2 *Indirect Systems.* The maximum permissible quantity of Group 2 *refrigerant* in an *indirect system* is shown in Table 3. Any system using a Group 2 *refrigerant* in excess of the quantities shown in Table 2 *shall* be of an *indirect* type as follows:

a) Institutional and Public Assembly Occupancies—*Indirect vented closed-surface,* o *double indirect vented open-spray.*

b) Residential and Commercial Occupancies — *Indirect vented closed-surface,* or *doubl indirect vented open-spray,* or *primary circuit of double-refrigerant* type.

6.3.2.1 *Machinery Rooms for Indirect Systems Group 2 Refrigerants*

6.3.2.1.1 *Indirect systems* using Group 2 *refrigerants* not in excess of the quantities shown i Column 1 of Table 3 *shall* have all refrigerant-containing parts, excepting parts mounted outside th building, installed in a *machinery room* used for n other purpose than for mechanical equipment.

TABLE 3. Maximum Permissible Quantities of Group Refrigerants for Indirect Systems

Occupancy	*Column 1* *Machinery Rooms (see 2.36), max lb*	*Column 2* *Class T Machinery Rooms (see 2.37), max lb*
Institutional	0	Not more than 500 l
Public Assembly	0	Not more than 1000 l
Residential	Not more than 300 lb	No limit
Commercial	Not more than 600 lb	No limit

6.3.2.1.2 *Indirect systems* using Group 2 *refrigerants* not in excess of the quantities shown i Column 2 of Table 3 *shall* have all refrigerant containing parts installed in a *Class T machiner room.*

6.3.2.1.3 *Flame-Producing Devices, Hot Sur- es, and Electrical Equipment in Machinery oms.* Where a *machinery room* is required by code to house a *refrigerating system* contain- any Group 2 *refrigerant* other than sulphur xide, no flame-producing device or hot surface ve 800 F *shall* be permitted in such room and electrical equipment in the room *shall* conform the requirements of Hazardous Locations Class f the latest edition of the National Electrical de (see Appendix 7). The use of matches, ciga- te lighters, halide leak detectors, and similar ices *shall not* be considered a violation of this agraph.

Group 3 Refrigerants

.4.1 Group 3 *refrigerants shall not* be used in titutional, Public Assembly, Residential, or mmercial Occupancies except in laboratories for mmercial Occupancies. In such laboratory in- llations only *unit systems* containing not more n 6 pounds *shall* be used unless the number of rsons does not exceed one person per 100 square t of laboratory floor area, in which case the re- rements for Industrial Occupancy *shall* apply.

ection 7. REQUIREMENTS FOR INDUSTRIAL OCCUPANCIES

General. There *shall* be no restriction on the antity or kind of *refrigerant* used in an Indus- al Occupancy, except as specified in 7.2 and 8.10.

Number of Persons. When the number of persons ove the first floor exceeds one person per 100 uare feet of floor area, the requirements of Com- rcial Occupancies *shall* apply unless that portion the building containing more than one person r 100 square feet of floor area above the first or, together with its *entrances* and *exits,* be cut from the rest of the building by tight construc- n with self-closing, tight-fitting doors.

Section 8. INSTALLATION REQUIREMENTS

Foundations and Supports for *condensing units* or *npressor units shall* be of substantial and non- mbustible construction when more than 6 inches gh. (See 8.3.)

Moving Machinery *should* be guarded in accord- ce with accepted safety standards. (See Appen- ces 3 and 4.)

Clear Space adequate for inspection and servic- of *condensing units* or *compressor units shall* provided.

Condenser Units or Compressor Units with Enclo- *es shall* be readily accessible for servicing and pection.

Water Supply and Discharge Connections *should* made in accordance with accepted safety and alth standards. (See Appendix 5.)

.5.1 Discharge water lines *shall not* be directly nected to the waste or sewer system. The waste discharge from such equipment *shall* be over d above a trapped and vented plumbing fixture.

8.6 Illumination adequate for inspection and servicing of *condensing units* or *compressor units should* be provided. (See Appendix 6.)

8.7 Electrical Equipment and Wiring *shall* be installed in accordance with accepted safety standards. (See Appendix 7.)

8.8 Gas Fuel Devices and Equipment used with *refrigerating systems shall* be installed in accordance with accepted safety standards. (See Appendix 8.)

8.9 Open Flames. When the quantity of flammable *refrigerant* in any one *refrigerating system* exceeds the amount given in Table 4 for each 1000 cubic feet of room volume in which the system or any part thereof is installed, then no flame-producing device or hot surface above 800 F *shall* be permitted in such room and all electrical equipment in the room *shall* conform to the requirements of Hazardous Locations Class I of the latest edition of the National Electrical Code. (See Appendix 7.)

TABLE 4. **Maximum Permissible Quantities of Flammable Refrigerants**

Name	*Chemical formula*	*Maximum quantity in lb per 1000 cu ft of room volume*
Butane	C_4H_{10}	2½
Ethane	C_2H_6	2½
Ethyl chloride	C_2H_5Cl	6
Ethylene	C_2H_4	2
Isobutane	$(CH_3)_3CH$	2½
Methyl chloride	CH_3Cl	10
Methyl formate	$HCOOCH_3$	7
Propane	C_3H_8	2½

8.10 Flammable Refrigerants as listed in Table 4 *shall not* be used in a *refrigerating system* in a built-up area in excess of 1000 pounds unless *approved* by the authority having jurisdiction.

8.11 Machinery Room Requirements (see Definitions 2.36 and 2.37)

8.11.1 Each refrigerating *machinery room shall* be provided with tight-fitting door or doors and have no partitions or openings that will permit the passage of escaping *refrigerant* to other parts of the building.

8.11.2 Each refrigerating *machinery room shall* be provided with means for ventilation to the outer air. The ventilation *shall* consist of windows or doors opening to the outer air, of the size shown in Table 5, or of mechanical means capable of removing the air from the room in accordance with Table 5. The amount of ventilation for *refrigerant* removal purposes *shall* be determined by the *refrigerant* content of the largest system in the *machinery room.*

8.11.3 *Mechanical Ventilation,* when used, *shall* consist of one or more power-driven exhaust fans, which *shall* be capable of removing from the refrigerating *machinery room* the amount of air specified in Table 5. The inlet to the fan, or fans, or *air duct* connection *shall* be located near the refrigerating equipment. The outlet from the fan, or fans, or *air duct* connections *shall* terminate outside of the building in an *approved* manner. When *air ducts* are used either on the inlet or discharge side of the fan, or fans, they *shall* have an area not less than specified in Table 5. Provision *should* be made for the inlet of air to replace that being exhausted.

8.11.4 ***Class T Machinery Rooms*** in basements or sub-basements (see Definition 2.37) *shall* have adequate mechanical ventilation operating continuously, which may be considered as a part of the emergency ventilation required in 8.11.2.

8.12 Air Duct Systems of air conditioning equipment for human comfort using mechanical refrigeration *should* be installed in accordance with accepted safety standards. (See Appendix 9.)

TABLE 5. Minimum Air Duct Areas and Openings

	Weight of refrigerant in system, lb	*Mechanical discharge of air, cfm*	*Duct area, sq ft*	*Open window or door area, sq ft*
up to	20	150	¼	4
	50	250	⅓	6
	100	400	½	10
	150	550	⅔	12½
	200	680	⅔	14
	250	800	1	15
	300	900	1	17
	400	1,100	1¼	20
	500	1,275	1¼	22
	600	1,450	1½	24
	700	1,630	1½	26
	800	1,800	2	28
	900	1,950	2	30
	1,000	2,050	2	31
	1,250	2,250	2¼	33
	1,500	2,500	2¼	37
	1,750	2,700	2¼	38
	2,000	2,900	2¼	40
	2,500	3,300	2½	43
	3,000	3,700	3	48
	4,000	4,600	3¾	55
	5,000	5,500	4½	62
	6,000	6,300	5	68
	7,000	7,200	5½	74
	8,000	8,000	5¾	80
	9,000	8,700	6¼	85
	10,000	9,500	6½	90
	12,000	10,900	7	100
	14,000	12,200	7½	109
	16,000	13,300	7¾	118
	18,000	14,300	8	125
	20,000	15,200	8¼	130
	25,000	17,000	8¾	140
	30,000	18,200	9	145
	35,000	19,400	9¼	150
	40,000	20,500	9½	155
	45,000	21,500	9¾	160

Section 9. REFRIGERANT PIPING, VALVES, FITTINGS, AND RELATED PARTS

9.1 General. *Refrigerant piping*, valves, fittings, and related parts used in the construction and installation of *refrigerating systems shall* conform to
American Standard Code for Pressure Piping. (
Appendix 10.)

**9.2 Metal Enclosures or Pipe Ducts for Soft Cop
Tubing.** Rigid or flexible metal enclosures *shall*
provided for soft, annealed copper tubing used
refrigerant piping erected on the premises
containing other than Group 1 *refrigerants*.
enclosures *shall* be required for connections
tween *condensing unit* and the nearest riser b
provided such connections do not exceed 6 feet
length.

**9.3 Specific Minimum Requirements for Refriger
Pipe and Tubing**

9.3.1 Standard wall steel or wrought iron p
may be used for *design working pressures* not
ceeding 300 psig, provided lap-welded, electric
sistance welded, or seamless pipe is used for si
2 inches and larger, and extra strong wall pipe
used for liquid lines for sizes 1½ inches
smaller conforming to American Standard B36
1939. (See Appendix 11.)

9.3.2 Standard iron pipe size copper and red br
(not less than 80 percent copper) pipe and tub
may be used and *shall* conform to the dimensi
given in Tables II and IV of ASTM Specificat
B188-47T. (See Appendix 12.)

9.3.3 Watertube size hard copper tubing used
refrigerant piping erected on the premises *s*
conform to ASTM Specification B88-48. Types
or L (see Appendix 13), for dimensions and sp
fications. Copper tubing ¼ inch and ⅜ inch
side diameter having a minimum wall thickness
0.025 inch and 0.030 inch, respectively, *shall*
considered as meeting the requirements for
copper tubing.

9.3.4 Soft annealed copper tubing used for *refri
erant piping* erected on the premises *shall not*
used in sizes larger than ⅞ inch outside diame
(¾ inch nominal). It *shall* conform to AS
Specification B68-47 (see Appendix 14), and *s*
have wall thicknesses not less than contained
ASTM Specification B88-48 (see Appendix 13)
Type L. Copper tubing ¼ inch and ⅜ inch outs
diameter having a minimum wall thickness of 0.
inch and 0.030 inch, respectively, *shall* be cons
ered as meeting the requirements for this cop
tubing.

9.3.5 Soldered joints on copper tubing used
refrigerating systems containing Group 2 or Gr
3 *refrigerants shall* be made with solders or all
having melting points not less than 1000 F.

9.4 Joints and Refrigerant-Containing Parts in Air Du
Joints and all refrigerant-containing parts o
refrigerating system located in an *air duct* of
air conditioning system for human comfort s
be constructed to withstand, without leakage
temperature of 1000 F.

9.5 Exposure of Refrigerant Pipe Joints. *Refriger*
pipe joints *shall* be exposed to view for visual
spection, excepting when insulated or mechanic

protected, or when located in the cabinet of a *unit system*, or installed outside the building, or in a tight *pipe duct* or shaft vented to the outer air.

9.6 Stop Valves

9.6.1 *General Requirements.* All systems containing more than 50 pounds of a Group 1 *refrigerant* or 6 pounds of a Group 2 or 3 *refrigerant*, other than systems utilizing *nonpositive displacement compressors, shall* have *stop valves* installed as follows:

a) Each inlet of each *compressor, compressor unit*, or *condensing unit;*

b) Each discharge outlet of each *compressor, compressor unit*, or *condensing unit*, and of each *liquid receiver.*

9.6.2 *Systems Containing 100 Pounds or More of Refrigerant.* All systems containing 100 pounds or more of a *refrigerant*, other than systems utilizing *nonpositive displacement compressors, shall* have *stop valves*, in addition to those in 9.6.1, on each inlet of each *liquid receiver* and each branch liquid and suction line except that none *shall* be required on the inlet of a *receiver* in a *condensing unit* nor on the inlet of a *receiver* which is an integral part of a *condenser.*

9.6.3 *Stop valves* used with soft annealed copper tubing or hard drawn copper tubing ¾ inch nominal size or smaller *shall* be securely mounted, independent of tubing fastenings or supports.

9.6.4 *Stop valves shall* be suitably labelled if it is not obvious what they control. Numbers may be used to label the valves provided a key to the numbers is located near the valves.

9.7 Location of Refrigerant Piping

9.7.1 *Refrigerant piping* crossing an open space which affords passageway in any building *shall* be not less than 7½ feet above the floor unless against the ceiling of such space.

9.7.2 Free passageway *shall not* be obstructed by *refrigerant piping. Refrigerant piping shall not* be placed in any elevator, dumbwaiter, or other shaft containing a moving object, or in any shaft which has openings to living quarters or to main *exit hallways. Refrigerant piping shall not* be placed in public *hallways, lobbies*, or stairways, except that such *refrigerant piping* may pass across a public *hallway* if there are no joints in the section in the public *hallway*, and provided nonferrous tubing of 1 inch nominal diameter (1⅛ inch outside diameter) and smaller be contained in a rigid metal pipe.

Section 10. DESIGN AND CONSTRUCTION OF EQUIPMENT

(Also see Section 11 for *pressure vessels*)

10.1 General

10.1.1 Every part of a *refrigerating system*, with the exception of pressure gages and control mechanisms, *shall* be designed, constructed, and assembled to withstand a test pressure not less than the minimum *refrigerant* leak field test pressure specified in Table 6 without being stressed beyond one-third of its ultimate strength. (See 10.2.)

NOTE: This paragraph establishes a minimum *design working pressure* in terms of the field test pressure so that the minimum *refrigerant* leak field test pressure, specified in Table 6, can be safely applied. Rules governing *pressure-relief devices, pressure-limiting devices*, etc., *shall* be based on the *design working pressure* selected.

10.1.2 All materials used in the construction and installation of *refrigerating systems shall* be suitable for conveying the *refrigerant* used. No material *shall* be used that will deteriorate because of the *refrigerant*, or the oil, or the combination of both.

NOTE: Many *refrigerants* are corrosive to the usual materials when moisture or air, or both, are present and it is assumed in approving these materials that the system will be charged and operated in accordance with accepted practice, to prevent or minimize this corrosion.

10.1.3 *Aluminum, Zinc, or Magnesium shall not* be used in contact with methyl chloride in a *refrigerating system.* Magnesium alloys *shall not* be used in contact with any Freon *refrigerant.*

10.2 Minimum Test Pressures. Every refrigerant-containing part of every system, including pressure gages and control mechanisms, *shall* be tested and proved tight by the *manufacturer* at not less than the minimum *refrigerant* leak field test pressure, specified in Table 6. (See 10.3.)

10.3 Equipment Listed by an Approved Nationally Recognized Testing Laboratory having a follow-up inspection service *shall* be deemed as meeting the intent of the requirements of 10.1 and 10.2. (See 1.2 and 2.3.1.)

10.4 Pressure-Limiting Devices

10.4.1 *Pressure-limiting devices shall* be provided on all systems containing more than 20 pounds of *refrigerant* and operating above atmospheric pressure, and on all water cooled systems so constructed that the *compressor* or *generator* is capable of producing a pressure in excess of the test pressure; except water cooled *unit systems* containing not more than 3 pounds of a Group 1 *refrigerant*, providing the system will safely relieve the *refrigerant* in case of failure of the water supply or providing an overload device will stop the action of the *compressor* before the pressure exceeds one-fifth the ultimate strength of the system.

10.4.2 *Pressure-limiting devices shall* stop the action of the *pressure-imposing element* at a pressure not more than 90 percent of the *pressure-relief device* setting, or 90 percent of the *refrigerant* leak field test pressure actually applied, whichever is lower. (See 10.2 for minimum *refrigerant* leak field test pressures.)

10.4.3 *Pressure-limiting devices shall* be connected, with no intervening *stop valves*, between the *pressure-imposing element* and any *stop valve* on the discharge side.

10.5 Liquid Level Gage Glasses, except those of the bull's-eye or reflex type, *shall* have automatic closing shut-off valves, and such glasses *shall* be adequately protected against injury.

10.6 Dial of a Pressure Gage, when the gage is permanently installed on the *high side* of a *refrigerating system, shall* be graduated up to approximately double the operating pressure, but in no case less than 1.2 times the *design working pressure.*

10.7 Nameplate. Each separately sold *condensing unit* and each *compressor* or *compressor unit* sold for field assembly in a *refrigerating system shall* carry a nameplate marked with the *manufacturer's name,* nationally registered trademark or trade name, identification number, and the name of the *refrigerant* for which it is designed.

Section 11. REFRIGERANT-CONTAINING PRESSURE VESSELS

11.1 Refrigerant-Containing Pressure Vessels Exceeding 6 Inches in Diameter. Refrigerant-containing *pressure vessels,* except those having a maximum allowable internal or external working pressure 15 psig or less, irrespective of size, or having an inside diameter of 6 inches or less irrespective of pressure, *shall* comply with the rules of Section VIII of the 1949 Edition of the ASME Boiler Construction Code (see Appendix 15) covering the requirements for the design, fabrication, and inspection during construction of unfired *pressure vessels.*

11.1.1 *Code Inspection Exemption.* Unfired *pressure vessels* within the following volume and pressure limits are exempted from the inspection requirements of Paragraph U-65 of Section VIII of the 1949 Edition of the ASME Boiler Construction Code:

a) 5 cubic feet gross capacity or less designed for pressures not exceeding 250 psig;

b) 1½ cubic feet gross capacity or less designed for pressures exceeding 250 psig;

but such vessels *shall* otherwise comply with Section VIII of the 1949 Edition of the ASME Boiler Construction Code requirements, including the required stamping but omitting the code symbol. The foregoing limitations apply to each single vessel, and not to an assembly of vessels.

11.1.2 *Pressure Vessels Less Than 5 Cubic Feet Capacity.* Except as specified in 11.2, *pressure vessels* which are exempted in 11.1.1 from compliance with Paragraph U-65 of Section VIII of the 1949 Edition of the ASME Boiler Construction Code, meet the requirements of Section 11 of this code, provided:

a) The details of design and construction *shall* conform with Section VIII of the 1949 Edition of the ASME Boiler Construction Code;

b) The materials of construction *shall* comply with the provisions of Section VIII of the 1949 Edition of the ASME Boiler Construction Code;

c) Welding operators and welding processes *shall* be qualified as provided for in Section VIII of the 1949 Edition of the ASME Boiler Construction Code;

d) Each vessel *shall* be tested and stamped in accordance with the Boiler Construction Code with either pneumatic or hydrostatic pressure in accordance with Par. U-77 (e);

e) The certification in 11.1.1 can be met by keeping a production series record of material used and tests made. Certification for individual vessels, where required, may be made by abstracting data from the series record by affidavit.

11.2 Refrigerant-Containing Pressure Vessels Not Exceeding an Inside Diameter of 6 Inches, irrespective of pressure, *shall* be listed either individually or as part of refrigeration equipment, by *an approved nationally recognized testing laboratory* having a follow-up inspection service. Vessels not so listed *shall* be constructed according to 11.1.2. (See 1.2 and 2.3.1.)

11.3 Safety Devices. All *pressure vessels,* irrespective of size or pressure, shall be equipped with safety devices in accordance with the requirements of Section 13 of this code.

11.4 Standard Hydrostatic Tests. Fusion welded *pressure vessels shall* be tested in accordance with Section VIII of the 1949 Edition of the ASME Boiler Construction Code. (See also 11.4.1.)

11.4.1 *Pneumatic Tests.* Vessels for use in services which cannot tolerate the presence of a testing liquid and which cannot be readily dried, and the parts of which have been previously tested by hydrostatic pressure to not less than 1½ times the *design working pressure* of the vessel, may be given a pneumatic test as prescribed in Paragraph U-64(f), Section VIII of the 1949 Edition of the ASME Boiler Construction Code.

Section 12. RELIEF DEVICES IN GENERAL

12.1 General. Every *refrigerating system shall* be protected by a *pressure-relief device* unless so constructed that pressure due to fire conditions wil be safely relieved by *soldered joints,* lead gaskets, *fusible plugs,* or other parts of the system.

12.1.1 No *stop valve shall* be located between any automatic *pressure-relief device* or *fusible plug* and the part or parts of the system protected thereby except when the parallel relief devices mentioned i 13.2 are so arranged that only one can be rendere inoperative at a time for testing or repair purposes.

12.1.2 All *pressure-relief devices shall* be connected as nearly as practicable directly to th *pressure vessel* or other parts of the system protected thereby, above the liquid *refrigerant* leve and installed so that they are readily accessible fo inspection and repair and so that they cannot b readily rendered inoperative. *Fusible plugs* ma be located above or below the liquid *refrigerar* level.

12.1.3 The seats and discs of *pressure-relief devices shall* be constructed of suitable material esist *refrigerant* corrosion or other chemical action caused by the *refrigerant.* Seats or discs of ast iron *shall not* be used.

12.1.4 The rated discharge capacity of a *pressure-elief valve* for a refrigerant-containing vessel, xpressed in pounds of air per minute, *shall* be etermined at a pressure at the inlet of the relief alve equal to 110 percent of the valve setting in ccordance with Paragraph UG-131, Section VIII of the 1950 Edition of the ASME Boiler Construction Code. (See Appendix 16.)

12.1.5 The rated discharge capacity of a *rupture member* or *fusible* plug in pounds of air per minute *shall* be determined by the following formulas:

$$C = 0.6\, P_1 d^2 \qquad (1)$$

or

$$d = 1.29 \sqrt{\frac{C}{P_1}}$$

where C = minimum required discharge capacity, in lb of air per min
d = minimum diameter of bore of *fusible plug* or internal diameter of inlet pipe to *rupture member* in inches

For *rupture members:*
P_1 = bursting pressure, equal to *design working pressure,* except that for 13.7 the bursting pressure equals 1.20 times *design working pressure,* psig

For *fusible plugs:*
P_1 = saturation pressure, corresponding to the stamped temperature melting point of the *fusible plug* or the critical pressure of the *refrigerant* used, whichever is smaller, psig

12.1.6 All *pressure-relief devices* (not *fusible plugs*) *shall* be directly pressure-actuated.

12.1.7 The size of the discharge pipe from the *pressure-relief device shall* be not less than the size of the relief device outlet. The discharge from more than one relief device may be run into a common header, the area of which *shall* be not less than the sum of the areas of the pipes connected thereto.

12.1.8 The length of discharge *piping* permitted to be installed on the outlet of a relief valve, *rupture member,* or *fusible plug shall* be determined as follows:

$$C = \frac{3\, P\, d^{5/2}}{L^{1/2}} \qquad (2)$$

or

$$d = \sqrt[5]{\frac{C^2 L}{9P^2}}$$

where C = minimum required discharge capacity in lb of air per min
d = internal diameter of pipe in in.
L = length of discharge pipe in ft
P = $0.25P_1$ (P_1 is defined under Equation 1.)

(See Table 7 for computations derived from the preceding formula.)

12.2 Pressure-Relief Devices for Positive Displacement Compressors. *Positive displacement compressors* operating above 15 pounds per square inch gage and having a displacement exceeding 50 cubic feet per minute, *shall* be equipped by the *manufacturer* with a *pressure-relief device* of adequate size and pressure setting to prevent rupture of the *compressor,* located between the *compressor* and *stop valve* on the discharge side. The discharge from such relief device may be vented to the atmosphere or into the low pressure side of the system.

12.3 Discharge of Pressure-Relief Devices and Fusible Plugs on all systems containing more than 6 pounds of *refrigerant,* other than systems containing Group 1 *refrigerants* in quantities less than shown in Table 1, Section 6, *shall* be to the outside of the building in an *approved* manner; *pressure-relief devices* may discharge into the *low side* of the system provided the relief devices are of a type not appreciably affected by back pressures and provided the *low side* of the system is equipped with relief devices vented to the outside of the building in an *approved* manner.

12.4 Ammonia Discharge. Where ammonia is used, the discharge may be into a tank of water which *shall* be used for no purpose except ammonia absorption. At least 1 gallon of fresh water *shall* be provided for each pound of ammonia in the system. The water used *shall* be prevented from freezing without the use of salt or chemicals. The tank *shall* be substantially constructed of not less than ⅛ inch or No. 11 U.S. gage iron or steel. No horizontal dimension of the tank *shall* be greater than one-half the height. The tank *shall* have hinged cover, or, if of the enclosed type, *shall* have a vent hole at the top. All pipe connections *shall* be through the top of the tank only. The discharge pipe from the *pressure-relief valves shall* discharge the ammonia in the center of the tank near the bottom.

12.5 Sulphur Dioxide Discharge. Where sulphur dioxide is used, the discharge may be into a tank of absorptive *brine* which *shall* be used for no purpose except sulphur dioxide absorption. There *shall* be 1 gallon of standard dichromate *brine* (2½ pounds sodium dichromate per gallon of water) for each pound of sulphur dioxide in the system. *Brines* made with caustic soda or soda ash may be used in place of sodium dichromate, provided the quantity and strength give the equivalent sulphur dioxide absorbing power. The tank *shall* be substantially constructed of not less than ⅛ inch or No. 11 U.S. gage iron or steel. The tank *shall* have a hinged cover, or, if of the enclosed type, *shall* have a vent hole at the top. All pipe connections *shall* be through the top of the tank only. The discharge pipe from the *pressure-relief valve shall* discharge the sulphur dioxide in the center of the tank near the bottom.

Section 13. RELIEF DEVICES FOR PRESSURE VESSELS

13.1 General. The rules of this section are based upon the rules given in Paragraphs UG-125 to UG-134, inclusive, of Section VIII of the 1950 Edition of the ASME Boiler Construction Code, with such additional restrictions as are necessary for control of *refrigerants.*

13.2 Pressure Vessels over 5 Cubic Feet. Each *pressure vessel* containing liquid *refrigerant* with internal gross volume (gross capacity) exceeding 5 cubic feet and exceeding 6-inch inside diameter, except as specified in 13.4, and which may be shut off by valves from all other parts of a *refrigerating system, shall* be protected by an *approved pressure-relief valve* in parallel with a *rupture member* or second *approved pressure-relief valve.* Each valve or *rupture member shall* be of sufficient capacity to prevent the pressure from rising more than 10 percent above the *design working pressure.* (See 12.1.1.)

13.2.1 *Relief Devices in Parallel on Large Vessels.* In cases where large *pressure vessels* containing liquid *refrigerant* except as specified in 13.4, require the use of two or more *pressure-relief devices* in parallel to obtain the capacity required by 13.5, the battery of *pressure-relief devices shall* be considered as a unit, and therefore as one *pressure-relief device.*

13.3 Pressure Vessels with Gross Volume of 5 Cubic Feet or Less. Each *pressure vessel* having a gross volume of 5 cubic feet or less, containing liquid *refrigerant,* except as specified in 13.4, and which may be shut off by valves from all other parts of a *refrigerating system, shall* be protected by a *pressure-relief device,* or *fusible plug.* This *shall not* apply to vessels having an inside diameter of 6 inches or less listed individually or as a part of refrigeration equipment by *an approved nationally recognized testing laboratory* having a follow-up inspection service. (See 1.2 and 2.3.1.) If a *fusible plug* is used, the ultimate bursting pressure of the vessed so protected *shall* be at least 2½ times the *refrigerant* saturation pressure, psig, corresponding to the stamped temperature on the *fusible plug,* or at least 2½ times the critical pressure of the *refrigerant* used, whichever is the smaller. A *fusible plug* is permitted only on the *high side* of a *refrigerating system.*

13.4 Relief Devices for Pressure Vessels Used as, or as Part of, Evaporator. Provisions of 13.2 and 13.2.1, which require a second parallel *approved* relief device, and 13.3 do not apply to *pressure vessels* used as, or as part of, *evaporators* insulated or installed in an insulated space, and which may be shut off by valves from all other parts of a *refrigerating system.*

13.5 Required Capacity. The minimum required rated discharge capacity of the *pressure-relief device* or *fusible plug* for a refrigerant-containing vessel *shall* be determined by the following formula:

$$C = fDL \qquad (3)$$

where C = minimum required discharge capacity of the relief device in lb or air per mir
D = outside diameter of the vessel in ft
L = length of the vessel in ft
f = factor dependent upon kind of *refrigerant,* as follows:

Kind of refrigerant	*Value of f*
Ammonia	0.5
Freon-12 and Freon-22	1.6
All other *refrigerants*	1.0

13.6 Pressure-Relief Device Setting. All *pressure-relie devices shall* be set to start to function at a pressure not to exceed the *design working pressure* o the vessel as determined by the *manufacturer* an stamped on the vessel or system.

13.7 Rupture Member Setting When Used in Paralle with Relief Valves. *Rupture members* used in paralle with *pressure-relief valves* on refrigerant-containing vessels *shall* function at a pressure not to exceed 20 percent above the *design working pressur* of the vessel.

13.8 Other Rupture Member Setting. All other *ruptur members* used in lieu of, or in series with, a relie valve *shall* function at a pressure not to exceed th *design working pressure* of the vessel and the conditions of application *shall* conform to the requirements of Section VIII of the 1950 Edition of th ASME Boiler Construction Code.

13.9 Marking of Relief Devices

13.9.1 All *pressure-relief valves* for refrigerant containing vessels *shall* be set and sealed by th *manufacturer.* Each relief valve *shall* be marke by the *manufacturer* with the data required i Paragraph UG-129 (a) of Section VIII of the 195 Edition of the ASME Boiler Construction Code.

13.9.2 Each *rupture member* for refrigerant containing *pressure vessels shall* be marked wit the information required in Paragraph UG-12 (d) of Section VIII of the 1950 Edition of th ASME Boiler Construction Code.

Section 14. FIELD TESTS

14.1 General. Every refrigerant-containing part every system that is erected on the premises, e cept *compressors, condensers, evaporators,* safe devices, pressure gages, and control mechanism that are factory tested, *shall* be tested and prov tight after complete installation, and before oper tion, at not less than the minimum *refrigerant* le field test pressures shown in Table 6.

14.2 Test Medium. No oxygen or any combustib gas or combustible mixture of gases *shall* be us within the system for testing.

14.3 Posting of Tests. A dated declaration of t *should* be provided for all systems containing lb or more of *refrigerant,* where required by 14 The declaration *should* be mounted in a fra protected by glass, and posted in the *machin room* and *should* give the name of the *refriger* and the field *refrigerant* leak test pressures

TABLE 6. Minimum Refrigerant Leak Field Test Pressures

		Minimum field refrigerant leak test pressures, psig	
igerant name	*Chemical formula*	*High pressure side*	*Low pressure side*
nonia	NH_3	300	150
ane	C_4H_{10}	95	50
oon dioxide	CO_2	1500	1000
llorodifluoromethane Freon-12)	CCl_2F_2	235	140
lloroethylene	$C_2H_2Cl_2$	30	30
lloromethane (Carrene No. 1) Methylene chloride)	CH_2Cl_2	30	30
lloromonofluoromethane Freon-21)	$CHCl_2F$	70	40
llorotetrafluoroethane Freon-114)	$C_2Cl_2F_4$	50	50
ane	C_2H_6	1200	700
yl chloride	C_2H_5Cl	60	50
ylene	C_2H_4	1600	1200
utane	$(CH_3)_3CH$	130	70
hyl chloride	CH_3Cl	210	120
hyl formate	$HCOOCH_3$	50	50
ochlordifluoromethane Freon-22)	$CHClF_2$	300	150
oane	C_3H_8	300	150
ohur dioxide	SO_2	170	85
hloromonofluoromethane Freon-11)	CCl_3F	30	30
hlorotrifluoroethane Freon-113)	$C_2Cl_3F_3$	30	30

TES:

For *refrigerants* not listed in Table 6 the test pressure for the high pressure side *shall* be not less than the aturated vapor pressure of the *refrigerant* at 150 F. The test pressure for the low pressure side *shall* be not ess than the saturated vapor pressure of the *refrigerant* at 110 F. In no case *shall* the test pressure be less than 30 psig.

When a *compressor* is used as a booster to obtain a low oressure and discharges into the suction line of another system, the booster *compressor* is considered a part of he *low side,* and values listed under the *low side* column n Table 6 *shall* be used for both *high* and *low side* of he booster *compressor* provided that a low pressure stage *compressor* of the *positive displacement* type *shall* nave a *pressure relief valve.*

In field testing systems using *nonpositive displacement compressors,* the entire system *shall* be considered for field test purposes as the *low side* pressure.

ed to the *high side* and the *low side* of the tem. The declaration of test *should* be signed the installer and, if an inspector is present the tests, he *should* also sign the declaration. en requested, copies of this declaration *shall* furnished to the enforcing authority.

Section 15. INSTRUCTIONS

Signs. Each *refrigerating system* erected on premises *shall* be provided with an easily legi- permanent sign securely attached and easily essible, indicating thereon the name and address the installer, the kind and total number of nds of *refrigerant* required in the system for normal operations, and the *refrigerant* leak field test pressure applied.

15.2 Metal Signs for Systems Containing More Than 100 Pounds of Refrigerant. Systems containing more than 100 pounds of *refrigerant shall* be provided with metal signs having letters not less than ½ inch in height designating the main shut-off valves to each vessel, main steam or electrical control, remote control switch, and *pressure-limiting device.* On all exposed high pressure and low pressure *piping* in each room where installed outside the *machinery room, shall* be signs, as specified above, with the name of the *refrigerant* and the letters "HP" or "LP."

15.3 New Sign for Changed Refrigerant. When the kind of *refrigerant* is changed as provided in 15.7 (Substitution of Refrigerant), there *shall* be a new sign, of the same type as specified in 15.2, indicating clearly that a substitution has been made, and stating the same information for the new *refrigerant* as was stated in the original.

15.4 Charging and Discharging Refrigerants. When *refrigerant* is added to a system, except a *unit system* requiring less than 6 pounds of *refrigerant,* it *shall* be charged into the low pressure side of the system. Any point on the downstream side of the main liquid line *stop valve shall* be considered as part of the low pressure side when operating with said *stop valve* in the closed position. No service *container shall* be left connected to a system except while charging or withdrawing *refrigerant.*

15.5 Refrigerants Withdrawn from Refrigerating Systems *shall* be transferred to *approved containers* only. (See Appendix 2.) No *refrigerant shall* be discharged to a sewer.

15.6 Containers Used for Refrigerants Withdrawn from a Refrigerating System *shall* be carefully weighed each time they are used for this purpose, and the *containers shall not* be filled in excess of the permissible filling weight for such *containers* and such *refrigerants* as are prescribed in the pertinent regulations of the Interstate Commerce Commission. (See Appendix 2.)

15.7 Substitution of Kind of Refrigerant in a system *shall not* be made without the permission of the approving authority, the user, and the makers of the original equipment, and due observance of safety requirements, including:

a) The effects of the substituted *refrigerant* on materials in the system;
b) The possibility of overloading the *liquid receiver* which *should not* be more than 80 percent full of liquid;
c) The liability of exceeding motor horsepower, *design working pressure,* or any other element that would violate any of the provisions of this code;
d) The proper size of *refrigerant* controls;
e) The effect on the operation and setting of safety devices;
f) The possible hazards created by mixture of the original and the substituted *refrigerant;*

g) Effect of the classification of the *refrigerant* as provided in this standard.

15.8 Refrigerant Stored in a Machinery Room *shall be* not more than 20 percent of the normal *refrigerant* charged nor more than 300 pounds of the *refrigerant,* in addition to the charge in the system and the *refrigerant* stored in a permanently attached *receiver,* and then only in *approved* storage *containers.* (See Appendix 2.)

15.9 Masks or Helmets. One mask or helmet *shall* be provided at a location convenient to the *machinery room* when an amount of a Group 2 *refrigerant* between 100 and 1000 pounds, inclusive, is employed. If more than 1000 pounds of a Group 2 *refrigerant* are employed, at least two masks or helmets *shall* be provided.

15.9.1 Only complete helmets or masks marked as *approved* by the Bureau of Mines of the United States Department of the Interior and suitable for the *refrigerant* employed *shall* be used and they *shall* be kept in a suitable cabinet immediately outside the *machinery room* or other *approved* accessible location.

15.9.2 Canisters or cartridges of helmets or masks *shall* be renewed immediately after having been used or the seal broken and, if unused, must be renewed at least once every two years. The date of filling *shall* be marked thereon.

15.10 Maintenance. All *refrigerating systems shall* be maintained by the user in a clean condition, free from accumulations of oily dirt, waste, and other debris, and *shall* be kept readily accessible at all times.

15.11 Responsibility as to Operation of the System. It *shall* be the duty of the person in charge of the premises on which a *refrigerating system* containing more than 50 pounds of *refrigerant* is installed, to place a card conspicuously as near as practicable to the *refrigerant compressor* giving directions for the operation of the system, including precautions to be observed in case of a breakdown or leak as follows:

a) Instruction for shutting down the system in case of emergency;
b) The name, address, and day and night telephone numbers for obtaining service;
c) The name, address, and telephone number of the municipal inspection department having jurisdiction, and instructions to notify said department immediately in case of emergency.

15.12 Pressure Gages *should* be checked for accuracy prior to an air test and immediately after every occasion of unusually high pressure, either by comparison with master gages or by setting the pointer as determined by a dead weight pressure gage tester.

TABLE 7. Length of Discharge Piping for Relief Va
or Rupture Members of Various Discharge Capac

Equiv. length of discharge pipe, ft (L)	*Discharge capacity in lb of air per min (C* *Standard wall iron pipe sizes, in.*						
	½	¾	1	1¼	1½	2	2½
			RELIEF DEVICE SET AT 25 PSIG (P_1)				
50	0.81	1.6	2.9	5.9	8.7	16.3	25.3
75	0.67	1.4	2.4	4.9	7.2	13.3	20.9
100	0.58	1.2	2.1	4.2	6.2	11.5	18.0
150	0.47	0.95	1.7	3.4	5.0	9.4	14.6
200	0.41	0.8	1.5	2.9	4.4	8.1	12.6
300	0.33	0.67	1.2	2.4	3.6	6.6	10.5
			RELIEF DEVICE SET AT 50 PSIG (P_1)				
50	1.6	3.3	5.9	11.9	17.4	32.5	50.6
75	1.3	2.7	4.9	9.7	14.3	26.5	41.8
100	1.2	2.3	4.2	8.4	12.3	23.0	36.0
150	0.94	1.9	3.5	6.9	10.0	18.7	29.2
200	0.81	1.6	2.9	5.9	8.7	16.3	25.3
300	0.66	1.3	2.5	4.9	7.1	13.3	21.0
			RELIEF DEVICE SET AT 75 PSIG (P_1)				
50	2.4	4.9	8.9	17.9	26.1	48.7	75.9
75	2.0	4.1	7.3	14.6	21.4	39.8	62.6
100	1.7	3.5	6.4	12.6	18.5	34.4	54.0
150	1.4	2.8	5.2	10.3	15.0	28.0	43.8
200	1.2	2.5	4.4	8.9	13.1	24.4	37.9
300	0.9	2.0	3.7	7.3	10.7	19.9	31.5
			RELIEF DEVICE SET AT 100 PSIG (P_1)				
50	3.2	6.6	11.9	23.8	34.8	65.0	101.2
75	2.7	5.4	9.7	19.4	28.6	53.0	83.6
100	2.3	4.6	8.5	16.8	24.6	45.9	72.0
150	1.9	3.8	6.9	13.7	20.0	37.4	58.4
200	1.6	3.3	5.9	11.9	17.5	32.5	50.6
300	1.3	2.7	4.9	9.7	14.2	26.5	42.0
			RELIEF DEVICE SET AT 150 PSIG (P_1)				
50	4.9	9.9	17.9	35.7	52.3	97.5	151.8
75	4.0	8.1	14.6	29.2	42.9	79.5	125.4
100	3.5	6.9	12.7	25.2	36.9	68.9	108.0
150	2.8	5.7	10.4	20.6	30.0	56.1	87.6
200	2.4	4.9	8.9	17.8	26.2	48.7	75.9
300	1.9	4.0	7.4	14.6	21.1	39.7	63.0
			RELIEF DEVICE SET AT 200 PSIG (P_1)				
50	6.5	13.2	23.8	47.6	69.7	130.0	202.4
75	5.3	10.8	19.4	38.9	57.2	106.0	167.2
100	4.6	9.2	16.9	33.6	49.2	91.8	144.0
150	3.8	7.6	13.8	27.4	40.0	74.8	116.8
200	3.2	6.5	11.8	23.8	34.9	64.9	101.2
300	2.6	5.3	9.8	19.4	28.4	52.9	84
			RELIEF DEVICE SET AT 250 PSIG (P_1)				
50	8.1	16.5	29.8	59.5	87.1	162.5	253.0
75	6.7	13.5	24.3	48.6	71.5	132.5	209.0
100	5.8	11.6	21.2	42.0	61.6	114.8	180.0
150	4.7	9.5	17.3	34.3	50.0	93.5	146.0
200	4.1	8.2	14.8	29.7	43.7	81.2	126.5
300	3.3	6.7	12.3	24.3	35.5	66.2	105.0
			RELIEF DEVICE SET AT 300 PSIG (P_1)				
50	9.7	19.8	35.7	71.4	104.5	195.0	303.6
75	7.9	16.2	29.1	58.3	85.8	159.0	250.8
100	6.9	13.9	25.4	50.4	73.9	137.7	216.0
150	5.6	11.3	20.7	41.1	60.0	112.2	175.2
200	4.9	9.8	17.8	35.6	52.4	97.4	151.8
300	3.9	7.9	14.7	29.1	42.6	79.4	126.0

APPENDIX

Titles of Reference Standards

is Appendix is not a part of the American Standard ty Code for Mechanical Refrigeration. This Appendix des complete titles and sources (indicated parentheti-) of various publications mentioned in the standard.

American Standard Method of Test for Flash Point by Means of the Pensky-Martens Closed Tester, ASTM D 93-46; API 510-46; ASA Z11.7-1947 (1)

Interstate Commerce Commission's Regulations for the Transportation by Rail of Explosives and other Dangerous Articles in Freight, Express, and Baggage Services including Specifications for Shipping Containers, effective October 1, 1930, with Supplement 18 effective May 8, 1948 (4)

American Standard Safety Code for Mechanical Power-Transmission Apparatus, B15-1927 (Reaffirmed 1935) (1)

Combustible Anesthetics in Hospital Operating Rooms —1944 (5)

American Standard Plumbing Code, A40.7-1949 (1)

American Recommended Practice of Industrial Lighting, A11-1942 (1)

National Electrical Code, C1-1946 (1)

National Board of Fire Underwriters Pamphlet 54—Installation, Maintenance, and Use of Piping and Fittings for City Gas, 1943 Edition (6)

National Board of Fire Underwriters Pamphlet 90—Air Conditioning, Warm Air Heating, Air Cooling, and Ventilating Systems, 1950 Edition (6)

American Standard Code for Pressure Piping, B31.1-1942, with Supplements B31.1a-1944 and B31.1b-1947 (1)

11. American Standard Wrought-Iron and Wrought-Steel Pipe, B36.10-1939 (1)
12. Specifications for Copper Bus Pipes and Tubes (Tentative), ASTM B188-47T (3)
13. Standard Specifications for Copper Water Tube, ASTM B88-49; ASA H23.1-1949 (1)
14. Standard Specifications for Seamless Copper Tubing, Bright Annealed, ASTM B 68-1947 (3)
15. Section VIII of the ASME Boiler Construction Code, 1949 Edition, familiarly known as ASME Unfired Pressure Vessel Code (7)
16. Section VIII of the ASME Boiler Construction Code, 1950 Edition, familiarly known as ASME Unfired Pressure Vessel Code (7)
17. For reference to items not found in the body of the code, the ASRE Data Book, Basic Volume, 1949, is recommended (2)

Reference Sources

(1) Obtainable from American Standards Association Incorporated, 70 East 45 Street, New York 17, N. Y.

(2) Obtainable from American Society of Refrigerating Engineers, 40 West 40 Street, New York 18, N. Y.

(3) Obtainable from American Society for Testing Materials, 1916 Race Street, Philadelphia 3, Pa.

(4) Obtainable from Interstate Commerce Commission, 12th Street and Constitution Avenue, Washington, D. C.

(5) Obtainable from National Fire Protection Association, 60 Batterymarch Street, Boston, Mass.

(6) Obtainable from National Board of Fire Underwriters, 85 John Street, New York 7, N. Y.

(7) Obtainable from American Society of Mechanical Engineers, 29 West 39 Street, New York 18, N. Y.

RINTED-2M-6/51

ASRE Standard 16-R

ASRE Standard

METHODS OF RATING AND TESTING AIR CONDITIONERS

Approved by the American Society of Refrigerating Engineers
DECEMBER 6, 1949

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40th STREET NEW YORK 18, N. Y.
Price $1.00

ASRE Committee on Rating and Testing Air Conditioners

D. D. Wile, *Chairman*	Refrigeration Engineering, Inc. (formerly with Carrier Corp.)
F. H. Faust, *Vice Chairman*	General Electric Company
H. A. Brysselbout	York Corp.
R. E. Holmes	Argonne National Laboratory (formerly with Westinghouse Electric Corp
J. Neuhoff	Worthington Pump & Machinery Corp.
H. C. Pierce	Servel, Inc.

Special acknowledgement is made to the following individuals who assisted the Committee and made valuable contributions to this Standard:

C. F. Schroeder	Servel, Inc.
I. P. Sharpe	General Electric Company

ASRE Standards Committee, 1948

R. C. Jordan, *Chairman*

D. D. Wile, *Vice Chairman*
R. E. Backstrom
J. R. Chamberlain
H. M. Hendrickson
G. B. Priester
M. F. Tokach

ASRE Standards Committee, 1949

Leon Buehler, Jr., *Chairman*

R. E. Backstrom
F. Y. Carter
J. R. Chamberlain
O. C. Eckel
H. M. Hendrickson
W. W. Higham
A. B. Newton

FOREWORD

This Standard on Rating and Testing Air Conditioners may be reproduced in whole or in part with due credit to The American Society of Refrigerating Engineers. Adherence is strictly on a voluntary basis and is in the interest of obtaining uniform standards in the industry.

The new Standard supersedes ASRE Standards 13 and 16. It includes the cooling and the heating and other functions of self-contained air conditioners, room air conditioners, remote-type air conditioners, heat pumps, and oil and gas fired air conditioners.

The test apparatus described in this Standard is basically of the same type as described in the original Standards 13 and 16 except for the inclusion of the Calibrated Room Calorimeter, which is primarily intended for tes room air conditioners of the free-delivery type. Since apparatus does not require attachments to air inlets outlets, it avoids the possibility of influencing the rati as might be the case where such attachments are app to certain types of room air conditioners.

Since the Calibrated Room Calorimeter is not now erally available for testing room air conditioners, i recommended that September 1, 1951, be set as the effe date for that portion of the Standard relating to delivery room air conditioners. It is recommended in the interim period, the Psychrometric Calorimete recognized as an alternate method of testing.

ASRE Standard

Methods of Rating and Testing Air Conditioners

CONTENTS

ection *Subject* *Page No.*

1 *Scope and Purposes*

1.10 Scope 3
1.20 Purposes 4
1.30 Method of Using This Standard 4

2 *Classification*

2.10 Methods of Rejecting Heat 5
2.20 Methods of Introducing Ventilation Air 5
2.30 Methods of Discharging Air to Room 5
2.40 Methods of Providing Refrigeration 5
2.50 Methods of Providing Heating 5
2.60 Location 5

3 *Basis of Rating*

3.10 Ratings 5
3.20 Standard Ratings 5
3.30 Application Ratings 5
3.40 Items Required in Rating 5
3.50 Published Ratings 6

4 *Instruments*

4.10 Temperature Measuring Instruments 6
4.20 Pressure Measuring Instruments 6
4.30 Air Capacity Measuring Instruments 6
4.40 Electrical Instruments 6
4.50 Volatile Refrigerant Flow Measuring Instruments 6
4.60 Brine or Water Flow Measuring Instruments 7
4.70 Speed Measuring Instruments 7
4.80 Specific Gravity Measuring Instruments 7
4.90 Other Measurements 7

5 *Apparatus*

5.00 Calorimeters and Auxiliary Apparatus 7
5.10 Psychrometric Calorimeter 7
5.20 Calibrated Room Calorimeter 8
5.30 Condensing Unit with Water Flow Measuring Apparatus 10
5.40 Calibrated Compressor 11
5.50 Air Flow and Temperature Measuring Apparatus 11
5.60 Liquid Flow Measuring Apparatus 12
5.70 Steam Condensate Measuring Apparatus..... 13

6 *Air Flow Measurement*

6.10 Apparatus 13
6.20 Nozzles 13
6.30 Calculation of Air Flow 13

7 *Liquid Flow Measurement*

7.10 Apparatus 14
7.20 Nonvolatile Refrigerant and Steam Condensate Flow Measurements 14

Section *Subject* *Page No.*

7.30 Volatile Refrigerant Flow Measurement in Liquid Phase, Using Calibrated Pressure Vessels 14
7.40 Volatile Refrigerant Flow Measurement in Liquid Phase, Using Liquid Quantity Meter 14
7.50 Volatile Refrigerant Flow Measurement, Using Water Cooled Condenser as Flow Meter.... 14
7.60 Volatile Refrigerant Flow Measurement, Using Calibrated Compressor 14

8 *Cooling Capacity Tests*

8.00 Requirement for Rating 15
8.10 Room Air Conditioners, Self-Contained, with Air or Evaporatively Cooled Condensers.... 16
8.20 Room Air Conditioners, Self-Contained, with Water Cooled Condensers 16
8.30 Air Conditioners, Self-Contained, with Air or Evaporatively Cooled Condensers 17
8.40 Air Conditioners, Self-Contained, with Water Cooled Condensers 18
8.50 Room Air Conditioners, Remote Type (for Use inside Conditioned Space) 19
8.60 Air Conditioners, Remote Type, Intended for Use within or Partly within Conditioned Space 19
8.70 Air Conditioners, Remote Type, Intended for Use outside Conditioned Space 19
8.80 Self-Contained Air Conditioners, Absorption Type 20

9 *Heating Capacity Tests*

9.00 Requirement for Rating 20
9.10 Room Air Conditioners 21
9.20 Air Conditioners for Use within Air Conditioned Room 21
9.30 Air Conditioners for Use outside of Conditioned Space 22
9.40 Air Conditioner Where Steam Is Used As Heating Medium 22
9.50 Air Conditioner Using Water As Heating Medium 22
9.60 Air Conditioner Using Electric Energy As Heating Medium 22
9.70 Air Conditioners, Direct Fired 22

10 *Humidifying Capacity Tests*

10.00 Requirement for Rating 22
10.10 Calculations 23

11 *Definitions*

12 *Symbols*

12.10 Primary Symbols 24
12.20 Subscript Symbols 24

Section 1. SCOPE AND PURPOSES

.10 *Scope*

1.11 This standard prescribes the methods of rating and testing all types of air conditioners which operate nonfrosting when cooling and dehumidifying at Standard Rating Conditions.

1.12 Various methods of classifying air conditioners are stated in Section 2 to illustrate but not to limit the scope of this standard.

1.13 Certain types of equipment which involve combustion of fuel, intermittent cycles, regeneration, recovery of energy, etc., may not be completely covered by this standard. However, the methods of rating, Standard Rating Conditions, and measurements can be used for portions of these types of equipment.

1.14 This standard does not include "Forced Circulation and Natural Convection Air Coolers for Refrigeration," for which see ASRE Standard 25.

1.20 *Purposes*

1.21 The purposes of this standard are to:

a. List and define the terms used in rating

b. Specify data required and establish conditions for obtaining Standard Ratings

c. Specify the type of data which should be included in Application Ratings

in obtaining

conditions

1.22 It is not the purpose of this standard to:

a. Provide the type of tests used for production field testing

b. Specify methods of calculating ratings for co ditions other than test conditions

1.30 *Method of Using This Standard*

1.31 Determine whether this standard is applicab by review of Sections 1, 2, and 11.

1.32 Select the Standard Rating Conditions fro Section 3.

Table 1. Standard Rating Conditions

Standard Barometric Pressure 29.92 in. Hg[1]

Type of Equipment to which rating applies	*Type of Application or Installation.*	*Description*	*Rating Condition*
		COOLING AND DEHUMIDIFYING	
All types	Self contained and remote units	Inlet temp. of air to be conditioned:	
	Comfort range	—dry bulb	80F
		—wet bulb	67F
	Process range	—dry bulb	65F
		—wet bulb	60F
	Food preservation range	—dry bulb	45F
		—wet bulb	42.9F
Air conditioners complete units or complete systems	Condenser Water cooled — self-contained units	Cooling water temp entering unit	75F
		Cooling water temp leaving unit	95F
	Air cooled and evaporatively cooled—self-contained units	Condenser air temp entering unit —dry bulb	95F
		—wet bulb	75F
	Remote—non-self-contained units	Saturated refrigerant vapor temperature leaving compressor	105F
		Refrigerant temperature entering liquid control device	95F minimum
Heat pumps complete units or complete systems	Well water cooled	Condenser water temp leaving unit	55F
		Condenser water temp leaving unit	75F
	Buried water coil	Condenser water temp entering unit	62F
		Condenser water temp leaving unit	68F
Air conditioner, Non-self-contained cooling units or cooling coils	Volatile refrigerant coil	Refrigerant temp entering liquid control device	95F
		Refrigerant temp corresponding to pressure of saturated vapor at outlet of evaporator or heat exchanger when provided[2]	40F
		Temperature of superheated refrigerant vapor at outlet of evaporator or heat exchanger when provided[2]	45F minimum
	Non-volatile refrigerant coil	Refrigerant temp entering cooling coil	40F
		Refrigerant temp leaving cooling coil	50F
		HEATING	
All types	All	Inlet temp of air to be heated	70F
All types except heat pump	Closed steam system	Dry saturated steam pressure	16.7 psia
	Atmospheric steam system	Dry saturated steam pressure[3]	14.7 psia
	Water circulating system	Water temp entering unit	180F
		Water temp leaving unit	160F
Heat pumps complete units or complete systems	Heat source from outside air	Outside air temp entering unit:	
		—dry bulb	20, 45F[4]
		—wet bulb	18.7, 42.9F[4]
	Heat source from well water	Water temp entering unit	50F
		Water temp leaving unit	43F
	Heat source from buried water coil	Water temp entering unit	40F
		Water temp leaving unit	37F
	All other heat sources	Refrigerant temp corresponding to pressure of saturated refrigerant vapor at outlet of evaporator or heat exchanger, when provided	30F
		Temperature of superheated refrigerant vapor at outlet of evaporator or heat exchanger when provided[2]	35F minimun
		HUMIDIFYING	
All types which humidify		Air-inlet dry bulb temperature	70F
		Air inlet wet bulb temperature	53F
		AIR CIRCULATION	
All types		Filters	new and clea

---

1. Method of correcting standard ratings to Standard Barometric Pressure of 29.92 in. Hg is specified in Section 6.
2. Refrigerant vapor temperature shall be the minimum at which no liquid refrigerant floods through at any time but not lower than temperature stated in table.
3. Dry saturated steam will be considered as steam having a minimum of 5 deg F and a maximum of 20 deg F superheat.
4. Ratings at both conditions required.

1.33 Select the apparatus required for test from Section 5 and the instruments from Section 4.

1.34 Test and calculate ratings in accordance with the appropriate methods in Sections 6, 7, 8, 9, and 10.

Section 2. CLASSIFICATION

.10 *Methods of Rejecting Heat*

2.11 Water cooled condenser
2.12 Air cooled condenser
2.13 Evaporatively cooled condenser

.20 *Methods of Introducing Ventilation Air*

2.21 Unit provides no ventilation air
2.22 Unit draws in outside air and discharges it to the room
2.23 Unit exhausts room air to the outside
2.24 Unit provides ventilation by a combination of methods 2.22 and 2.23.

.30 *Methods of Discharging Air to Room*

2.31 Free delivery
2.32 Through ducts

.40 *Methods of Providing Refrigeration*

2.41 Mechanical compression
2.42 Absorption
2.43 Water and brine

.50 *Methods of Providing Heating*

2.51 Steam
2.52 Hot water
2.53 Heat pump
2.54 Electrical
2.55 Direct fired

.60 *Location*

2.61 Within the conditioned space
2.62 Outside of the conditioned space
2.63 Partly within the conditioned space

Section 3. BASIS OF RATING

.10 *Ratings*

3.11 Ratings of air conditioners shall consist of:

a. Quantitative effects produced upon air such as cooling and dehumidifying in Btu per hr, heating in Btu per hr, humidifying in lb per hr, and air flow in cfm under specified conditions.

b. Other data pertaining to the application of the equipment such as power input in watts or kilowatts, water consumption in gal per hr.

3.12 Unless otherwise specified (as in Pars. 8.02, 9.01, and 10.01) the average of two methods of determining the ratings simultaneously, the results of which agree within 6.0 percent, is required for the principal ratings of cooling and heating capacity.

3.13 It should be noted that agreement of two test results within 6.0 percent does not in itself insure the accuracy of a rating even though a perfect heat balance is attained. The rating will be in error if the temperature and humidity of air entering the air conditioner or if other temperatures as specified in Table 1 are improperly measured. In order to insure accurate ratings, all readings must be accurately made within the variations allowed in this standard.

Table 2. Variations Allowed in Test Readings from Rating Conditions Specified in Table 1.

Reading	*Variation of arithmetical average*	*Maximum variation of individual readings*
COOLING AND DEHUMIDIFYING		
All entering air temperatures:		
Dry bulb	0.5 deg F	2.0 deg F
Wet bulb	0.3 deg F	1.0 deg F
Saturated suction temperature	0.5 deg F	3.0 deg F
Liquid flow temperature	0.2 deg F	0.5 deg F
Cooling water entering and leaving temperatures	0.2 deg F	
Voltage	2 %	
HEATING AND HUMIDIFYING		
All air temperatures:		
Dry bulb	0.5 deg F	2.0 deg F
Wet bulb	1.0 deg F	2.0 deg F
Steam pressure	0.5 psi	1.0 psi
Liquid flow temperature	0.2 deg F	0.5 deg F
Voltage	2 %	
ALL		
Liquid flow rates		2.0 %

3.20 *Standard Ratings*

3.21 Standard Ratings of air conditioners shall be measured under the Standard Rating Conditions specified in Table 1.

3.22 Variations allowed in test readings from Standard Rating Conditions are specified in Table 2.

3.23 Test results shall be used as Standard Ratings without adjustment for variations allowed in test readings (Table 2) except as specified to compensate for deviations from standard barometric pressure.

3.30 *Application Ratings*

3.31 Application Ratings are those covering performance under various conditions of operation or application for which the air conditioner is suitable.

3.32 Test procedures and measurements prescribed in this code may be used for Application Ratings. The conditions of operation should be specifically stated.

3.40 *Items Required in Rating*

3.41 Functions which unit performs, such as:

a. Cooling
b. Dehumidifying
c. Heating
d. Humidifying
e. Air circulating
f. Ventilating
g. Filtering
h. Air washing

3.42 Other information where applicable:

a. Manufacturer's name and address
b. Model, size, or type
c. A listing of all rating conditions specified in Table 1 that are applicable
d. Net total cooling effect, Btu per hr
e. Net dehumidifying effect, Btu per hr
f. Net sensible cooling effect, Btu per hr
g. Net sensible heating effect, Btu per hr
h. Humidifying effect, lb of water per hr
i. Total air capacity for heating and cooling, cfm of Standard Air
j. Ventilation air capacity, in cfm of Standard Air, and information as to whether provided by sup-

plying outdoor air, exhausting room air, or a combination of both
k. Name of refrigerant
l. Refrigerant saturated suction temperature corresponding to pressure and superheat temperature at outlet of the unit, deg F (not required for self-contained units)
m. Liquid refrigerant temperature entering unit (not required on self-contained units)
n. Cooling water flow rate, gal per hr
o. Quantity of nonvolatile refrigerant circulated per minute (for non-self-contained units only), lb per min
p. Pressure drop through coil when using nonvolatile refrigerant (for non-self-contained units only), lb per sq in.
q. Pressure drops through coil circuits and heat exchanger circuits using volatile refrigerant (for non-self-contained units only), lb per sq in.
r. Power input of each motor separately stated, watts
s. Total power input, watts
t. Nameplate ratings of each motor
u. Heat input, Btu per hr
v. Spray pressure, lb per sq in.
w. Total water consumption for humidifying, lb per hr
x. Pressure available for resistance of duct system, in. of water
y. Manufacturer's instructions and requirements which may affect performance

3.50 *Published Ratings*

3.51 Published Ratings, in order to conform with this standard, shall include Standard Ratings identified as such.

3.52 When published ratings cover more than one range, as stated in Table 1, the published ratings shall include the Standard Rating for each range.

3.53 Published ratings shall state whether the air conditioner is primarily intended for use inside or outside of the conditioned space and under which of these conditions the rating is based.

Section 4. INSTRUMENTS

4.10 *Temperature Measuring Instruments*

4.11 Temperature measurements shall be made with one or more of the following instruments:
a. Mercury-in-glass thermometers
b. Thermocouples
c. Electric resistance thermometers

4.12 Accuracy shall be within the following limits:
a. Wet and dry bulb temperatures, ± 0.10 deg F
b. Water and nonvolatile refrigerant temperatures, ± 0.10 F
c. All other temperatures, ± 0.50 deg F

4.13 In no case shall the smallest scale division of the temperature measuring instrument exceed twice the specified accuracy. For example, if the specified accuracy is ± 0.10 deg F, the smallest scale division shall not exceed 0.20 deg F.

4.14 Where an accuracy closer than ± 0.50 deg F is specified, the instrument shall be calibrated by comparison with a certified standard in the range of use or shall itself be certified as to accuracy.

4.15 Wet bulb temperatures shall be read only under conditions which assure an air velocity of 700 to 2000, preferably near 1000, ft per min over the wet bulb, and only after sufficient time has been allowed for evaporative equilibrium to be attained.

4.16 Wherever possible, temperature measuring
struments used to measure the change in tempe
ture of a liquid or a gas shall be arranged so t
they can be readily interchanged between i
and outlet positions after every reading, so as
improve accuracy.

4.17 Temperature of fluids within conduits shall
measured by inserting the temperature measur
instrument directly within the fluid or withi
well inserted into the conduit. Glass thermomet
should not be inserted directly into the fluid w
pressure within the conduit is great enough
affect the thermometer reading.

4.20 *Pressure Measuring Instruments*

4.21 Pressure measurements shall be made with
or more of the following instruments:
a. Mercury column
b. Bourdon tube gage

4.22 Accuracy of pressure measuring instrume
shall permit pressure measurements within ±
percent absolute.

4.23 Calibration of the Bourdon tube gage shall
with respect to a dead-weight tester or by comp
son with a mercury column.

4.24 In no case shall the smallest scale division
the pressure measuring instrument exceed two
one half times the specified accuracy.

4.30 *Air Capacity Measuring Instruments*

4.31 Static pressures across nozzles shall be measu
by means of manometers which have been
brated against a standard manometer to ±
percent of the reading. The smallest manom
scale division shall not exceed 2.0 percent of
reading.

4.32 Areas of nozzles shall be determined by m
uring their diameters to an accuracy of ±
percent in four places approximately 45 deg
apart around the nozzle in each of two pl
through the nozzle throat, one at the outlet
the other in the straight section near the radiu

4.40 *Electrical Instruments*

4.41 Electrical measurements shall be made with
or more of the following instruments:
a. Indicating
b. Integrating

4.42 Accuracy shall be within the following limit
a. Instruments used for measuring the elect
input to heaters or other accessories furnis
sensible or latent heat loads shall be accurat
± 1.0 percent of the quantity measured.
b. Instruments used for measuring the elect
input to fan motors, compressor motors, p
motors, or other accessories shall be accurat
± 2.0 percent of the quantity measured.

4.50 *Volatile Refrigerant Flow Measuring Instruments*

4.51 Flow measurements shall be made with on
more of the following instruments or methods
ing an accuracy of ± 2.0 percent of the qua
measured:
a. Liquid refrigerant quantity meter, measu
either weight or volume
b. Liquid refrigerant flow meter
c. Condenser water method
d. Calibrated compressor method

0 *Brine or Water Flow Measuring Instruments*

.61 Volume measurements shall be made with one or more of the following instruments having an accuracy of ± 1.0 percent of the quantity measured:

a. Liquid quantity meter, measuring either weight or volume

b. Liquid flow meter

70 *Speed Measuring Instruments*

.71 Speed measurements shall be made with one or more of the following instruments having an accuracy of ± 1.0 percent:

a. Revolution counter

b. Tachometer

c. Stroboscope

d. Oscilloscope

80 *Specific Gravity Measuring Instruments*

4.81 Brine specific gravity shall be read with a hydrometer or other means accurate within ± 1.0 percent of the specific gravity read.

90 *Other Measurements*

4.91 Time measurements shall be made with instruments whose accuracy is ± 0.20 percent.

4.92 Weight measurement shall be made with apparatus whose accuracy is ± 0.20 percent.

Section 5. APPARATUS

00 ***Calorimeters and Auxiliary Apparatus*** required for testing various types of air conditioners.

5.01 *Room Air Conditioners, Self-Contained, with Air or Evaporatively Cooled Condensers* of the type installed in or in front of a window, shall use the following apparatus:

a. Calibrated Room Calorimeter (Sec. 5.20) for both conditioned air and condenser air sides.

5.02 *Room Air Conditioners, Self-Contained, with Water Cooled Condensers* shall use the following apparatus:

a. Calibrated Room Calorimeter (Sec. 5.20) for conditioned air side.

b. Liquid Flow Measuring Apparatus (Sec. 5.60) for condenser water flow measurement.

5.03 *Air Conditioners, Self-Contained, with Air or Evaporatively Cooled Condensers*, when intended for use within or partly within the conditioned space, shall use the following apparatus:

a. Calibrated Room Calorimeter (Sec 5.20) for both conditioned air and condenser air sides, or

b. Psychrometric Calorimeter (Sec. 5.10) with Enclosure for both conditioned and condenser air sides.

5.04 *Air Conditioners, Self-Contained, with Water Cooled Condensers* shall use the following:

a. Psychrometric Calorimeter (Sec. 5.10) when conditioner is intended for use within, partly within, or outside the conditioned space, or

b. Calibrated Room Calorimeter (Sec. 5.20) when conditioner is intended for use within the conditioned space.

c. Liquid Flow Measuring Apparatus when conditioner is intended for use within, partly within or outside the conditioned space, or for condenser water flow measurement (Sec. 5.60).

5.05 *Room Air Conditioners, Remote Compressor or Condenser Type*, when intended for use within or partly within the conditioned space, shall use the following apparatus:

a. Calibrated Room Calorimeter (Sec. 5.20) for the conditioned air side.

b. Condensing Unit with Water Flow Measuring Apparatus (Sec. 5.30) for the refrigerant side, or

c. Calibrated Compressor (Sec. 5.40) for the re-refrigerant side, or

d. Liquid Flow Measuring Apparatus (Sec. 5.60) for volatile refrigerant for the refrigerant side, or

e. Liquid Flow Measuring Apparatus (Sec. 5.60) for nonvolatile refrigerant for the refrigerant side.

5.06 *Air Conditioners, Remote Compressor or Condenser Type*, intended for use within, partly within, or outside the conditioned space, shall use the following apparatus:

a. Psychrometric Calorimeter (Sec. 5.10) or Calibrated Room Calorimeter (Sec. 5.20) for the conditioned air side.

b. Condensing Unit with Water Flow Measuring Apparatus (Sec. 5.30) for the refrigerant side, or

c. Calibrated Compressor (Sec. 5.40) for the refrigerant side, or

d. Liquid Flow Measuring Apparatus (Sec. 5.60) for volatile refrigerant for the refrigerant side, or

e. Liquid Flow Measuring Apparatus (Sec. 5.60) for nonvolatile refrigerant for the refrigerant side.

5.07 *Air Conditioners, Absorption Type*, shall use the following apparatus:

a. Psychrometric Calorimeter (Sec. 5.10).

b. Liquid Flow Measuring Apparatus (Sec. 5.60).

c. Steam Condensate Measuring Apparatus (Sec. 5.70), or other suitable apparatus to determine heat input to absorption unit.

5.08 *Air Conditioners, All Types Including Heat Pumps, Heating and Humidifying Functions* shall use the following apparatus:

a. Air side measurements of heating and humidifying capacity may be made with calorimeters as specified in Paragraphs 5.01 through 5.07 for the particular type of air conditioner. Room air conditioners tested in a calibrated room calorimeter may be installed in the outdoor compartment so that the heating and humidifying capacity can be absorbed by the heat and moisture removing equipment of the calorimeter.

b. Steam heating capacity, as determined by measurement of the heating medium, shall be measured by apparatus described in Section 5.70.

c. Hot water heating capacity, as determined by measurement of the heating medium, shall be measured by apparatus described in Section 5.60.

5.10 *Psychrometric Calorimeter (Figs. 1, 2)*

5.11 This calorimeter determines the change in heat content of conditioned air by the measurement of (a) air flow, and (b) dry and wet bulb temperatures of incoming and outgoing air. The enclosure which forms a part of the calorimeter permits measurement of jacket losses. If the air conditioner is intended for installation within the conditioned space, jacket losses are included in the capacity measurement by establishing specified inlet air conditions at the inlet to the enclosure. If the air conditioner is intended for installation outside the conditioned space, jacket losses are used to determine net refrigerating effect on the refrigerant side but are excluded from the air side capacity measurements by establishing specified inlet air conditions at the inlet to the air conditioner.

5.12 The calorimeter shall include air flow measuring apparatus as described in Section 5.50 and

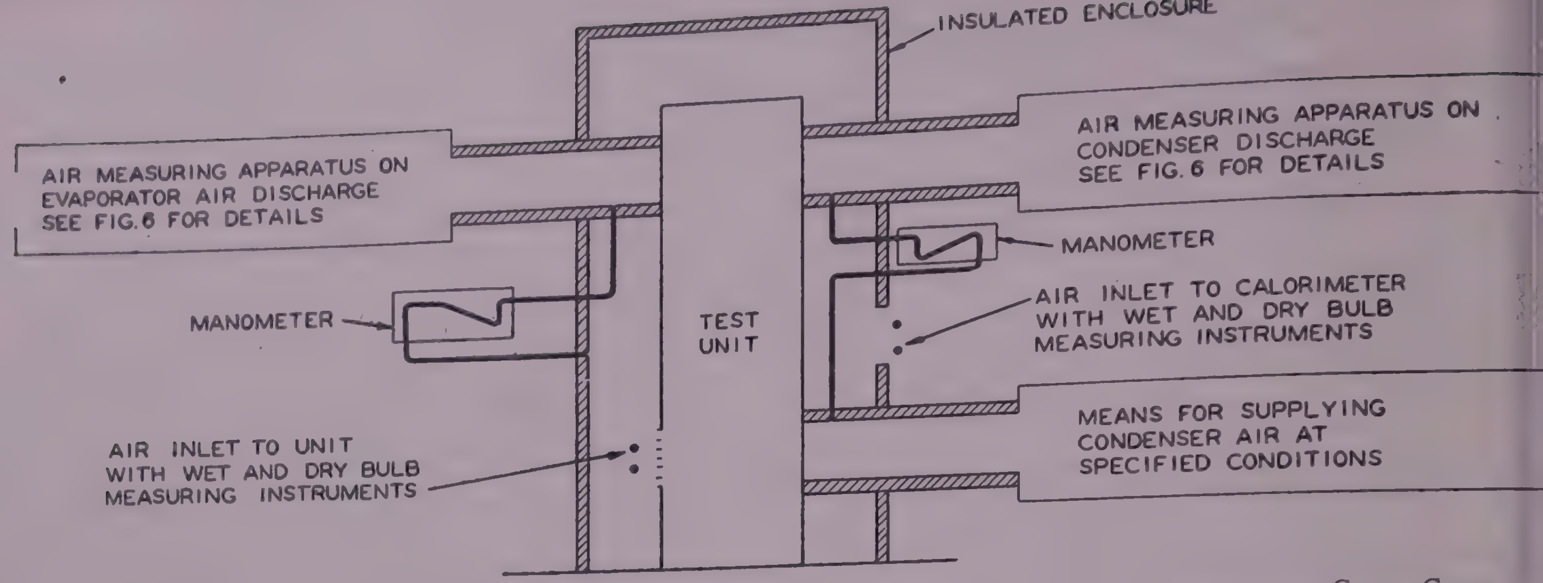

INSULATED ENCLOSURE
AIR MEASURING APPARATUS ON CONDENSER DISCHARGE SEE FIG. 6 FOR DETAILS
AIR MEASURING APPARATUS ON EVAPORATOR AIR DISCHARGE SEE FIG. 6 FOR DETAILS
MANOMETER
AIR INLET TO CALORIMETER WITH WET AND DRY BULB MEASURING INSTRUMENTS
MANOMETER
TEST UNIT
MEANS FOR SUPPLYING CONDENSER AIR AT SPECIFIED CONDITIONS
AIR INLET TO UNIT WITH WET AND DRY BULB MEASURING INSTRUMENTS

Fig. 1. Psychrometric Calorimeter Showing Air Measuring and Supply Apparatus for Air or Evaporatively Cooled Condense

shown in Figure 6. It need not be sealed but shall be essentially airtight. Openings to accommodate water, electrical, and other connections shall be of such size as to minimize air leakage from the test room to the inside of the calorimeter. It may be constructed of any suitable material and should preferably be insulated. A means for access to the test unit may be provided in the enclosure.

5.13 Size of the enclosure shall be sufficient to permit circulation of the inlet air between the enclosure and the air conditioner without excessive restriction, but in no case shall the distance between the sides of the air conditioner and the inside of the enclosure be less than six inches. The inlet air opening in the enclosure, as shown in Figure 1, should be located in the wall opposite to the inlet of the air conditioner in order to cause circulation throughout the enclosure.

5.14 Flow of conditioned air shall be measured by apparatus, described in Section 5.50, connected to the discharge opening of the air conditioner. For air cooled or evaporatively cooled conditioners a source of controlled air, maintained at conditions specified in Table 1, shall be connected to the outside air intake. The outside air discharge shall be connected to an air flow measuring apparatus as described in Section 5.50. All connecting ducts shall be sealed to prevent air leakage and insulated to prevent excessive heat leakage.

5.15 Manometers, as shown in Figures 1 and 2, shall be connected between the intake and discharge connections of the conditioned air circuit and of th outside air circuit, when used, to indicate th total pressure difference between intake and di charge. During operation the apparatus shall b adjusted to maintain the manometer readings a zero plus or minus 0.005 in. of water.

5.16 Air velocity over wet bulb thermometers shall b not less than 700 ft per min. If such velocities ar not available, sampling tubes as described in Section 5.20 shall be used. Uniformity of temperature entering an opening shall be checked by n less than two wet and two dry bulb measurement spaced over equal areas. Accurate readings ar essential for accurate net ratings. It may be advantageous to use more than two wet and dry bul measurements in order to have a better basis fo determining whether readings are properly mad

5.17 When water flow measurements and water temperature measurements are required, the wate pipes shall be insulated at the point of temperature measurements and between that point an the inlet to or the outlet from the conditioner.

5.18 When cooling or heating is accomplished wit liquids or steam passing through the air conditione from an outside source, apparatus is required a described in Sections 5.60 and 5.70 respectively.

5.20 *Calibrated Room Calorimeter (Fig. 3)*

5.21 This calorimeter determines the capacity of a air conditioner by balancing the cooling and dehumidifying effects with a measured heat i

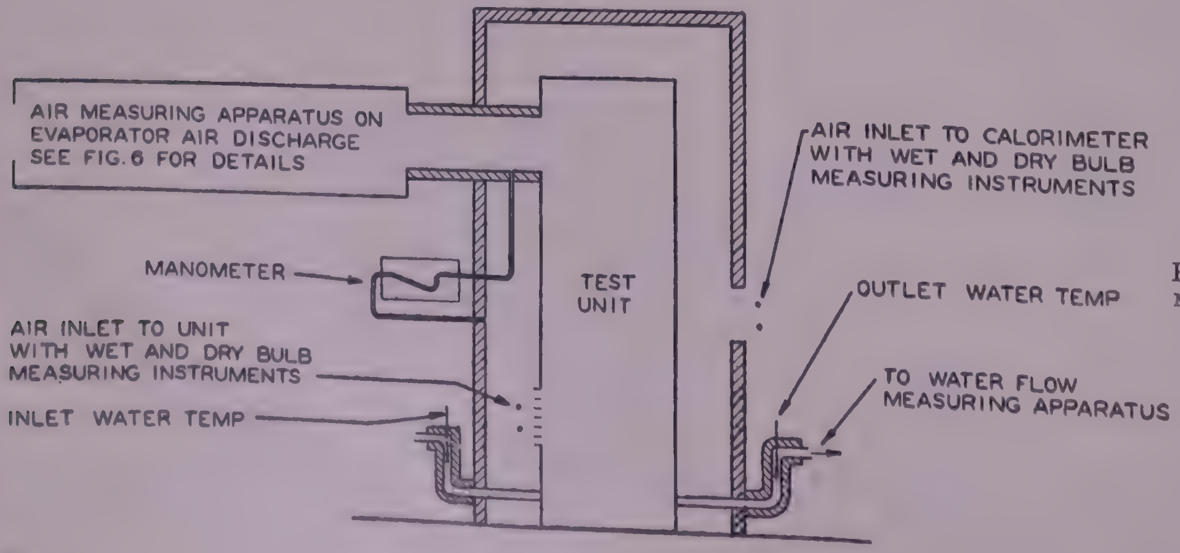
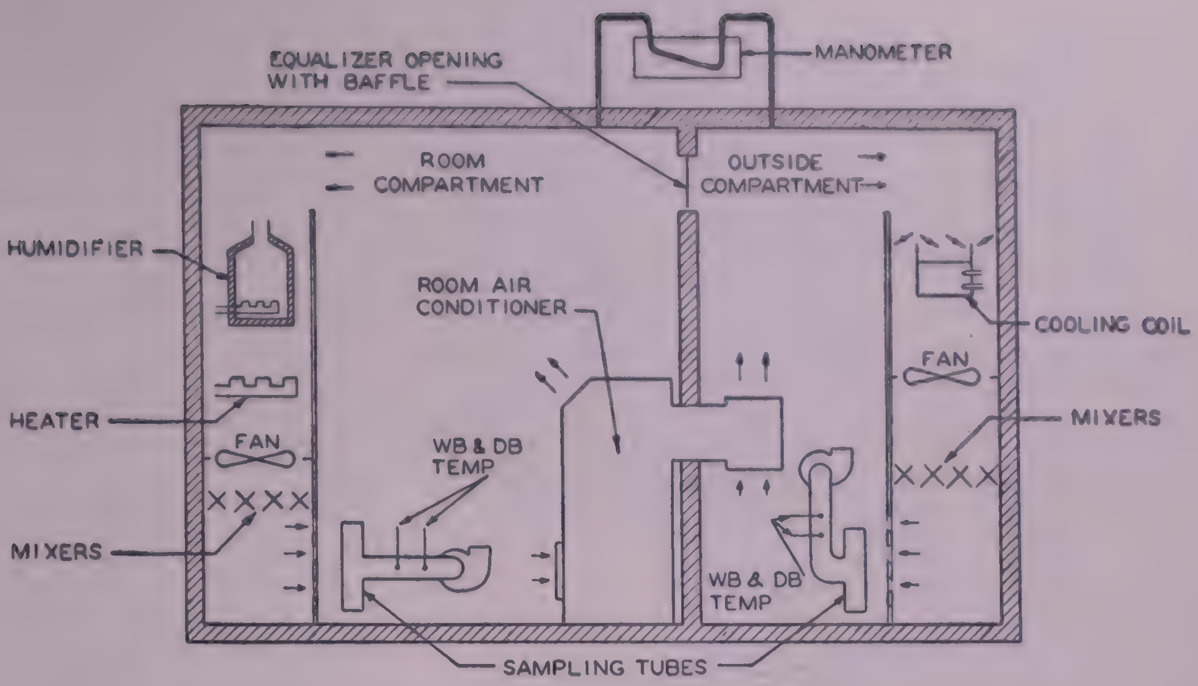

AIR MEASURING APPARATUS ON EVAPORATOR AIR DISCHARGE SEE FIG. 6 FOR DETAILS
AIR INLET TO CALORIMETER WITH WET AND DRY BULB MEASURING INSTRUMENTS
MANOMETER
TEST UNIT
OUTLET WATER TEMP
AIR INLET TO UNIT WITH WET AND DRY BULB MEASURING INSTRUMENTS
TO WATER FLOW MEASURING APPARATUS
INLET WATER TEMP

Fig. 2. Psychrometric Calor meter Showing Water Cool Condensing Unit.



3. Calibrated Room Calorimeter.

put. Heating and humidifying effects are balanced by a measured amount of cooling medium. The confirming test of the cooling and dehumidifying effects is obtained by balancing the heat rejected by the condenser with a measured amount of cooling medium. It is recognized that some types of room air conditioners may, in normal operation, recirculate a portion of the conditioned air into the return opening or, in the case of the condenser air, may recirculate a portion of the discharge air into the intake opening. For this reason room air temperature and outdoor air temperature are determined at some distance from the conditioner rather than at the conditioner intakes where recirculated air may be present. Furthermore, the design of the calorimeter is intended neither to augment nor retard any recirculation of air that may occur in normal operation. The quantity of air circulated by the room air conditioner does not affect measurement of capacity and is not measured directly but may be determined from the cooling capacity of the room air conditioner and the change of enthalpy of the air between entering and leaving conditions.

.22 This calorimeter, while intended primarily for free delivery fan air conditioners for use within or partly within the conditioned space and subject to recirculation of conditioned and condenser air, may be used for any air conditioner where it is desired to obtain net cooling, dehumidifying, and heating effects and net heat rejection. The left or right portion only of this calorimeter is required where recirculation and free delivery occur in either the conditioned or condenser air, respectively.

.23 The calibrated room calorimeter consists of two main compartments: the evaporator or room compartment and the condenser or outdoor compartment. These compartments are separated by an insulated partition having an opening similar to the one to which the air conditioner is normally connected. The air conditioner shall be installed at this opening using supporting members and filler pieces in a manner identical to a normal installation. No effort shall be made to seal the internal construction of the air conditioner to prevent air leakage from the condenser side to the evaporator side. No connections or alterations shall be made to the conditioner which might in any way interfere with its normal operation.

5.24 An opening shall be provided in the partition between the outside compartment and the room compartment to maintain a balanced pressure between these two compartments, thus freely permitting exchange of inside and outside air. This opening should have a damper made of vaporproof cloth or other similar light material to prevent circulation by gravity while at the same time permitting free movement of air from one compartment to the other. Size of the opening shall be such as to provide a free area of not less than one square foot for each 10,000 Btu per hr capacity of the unit under test, but in no case small enough to permit a static pressure difference between the two compartments in excess of 0.005 in. of water. This opening permits testing air conditions with ventilating dampers open since the capacity under these conditions can be correctly determined from both heat input to the room compartment and heat removal from the outdoor compartment. If desired, the amount of ventilation air may be measured by installing in the opening an air flow measuring device such as described in Section 5.50. This arrangement also permits measuring the amount of ventilation air when a static pressure difference exists between the room and the outside if the compartments are tightly sealed against air leakage. If such an air measuring device is used, the energy input to its fan should be small and should be included in the heat balance.

5.25 Size of the calorimeter shall be sufficient to avoid any restriction to intake or discharge openings of the air conditioner. Perforated plates or other suitable grilles shall be provided under the discharge openings from the air treating equipment to avoid face velocities exceeding 100 ft per min. Sufficient space should be allowed in front of any inlet or discharge grilles of the conditioner to avoid interference of the air flow. Minimum distance from the air conditioner to walls or ceiling of the calorimeter shall be three feet except for the back of the conditioner, which shall be in normal relation to the wall.

5.26 Each compartment shall be provided with air treating equipment to circulate the air and maintain prescribed conditions. Air treating equipment for the room compartment shall consist of heaters to supply the sensible heat and a humidifier to supply the moisture. The energy supply may be electric, steam, or any other source that can be readily controlled and measured. Air treating equipment for the outdoor compartment shall provide cooling and dehumidification. A cooling coil equipped with by-pass dampers to control the dry bulb temperature and supplied with variable temperature water or variable water quantity to control the wet bulb temperature may be used. If desired, dehumidifying apparatus or reheating apparatus or both may be used in combination with the cooling coil. Air treating equipment for both compartments shall be located behind suitable baffles and provided with fans of sufficient capacity to overcome the resistance of the air treating equipment and circulate not less than two times the quantity of air circulated by the air conditioner to the room or to the outdoors as the case may be. In no case shall the air treating equipment circulate less than one air change per minute to the compartments. The wet bulb temperature of air leaving the air treating equipment shall not differ from one point to another by more than one degree measured at the air treating equipment discharge. If necessary, air mixing devices as indicated in Figure 3 may be used. A suggested design for air mixing devices is described by D. D. Wile in "Air Flow Measurement in the Laboratory," (*Refrig. Eng.*, June 1947, page 515).

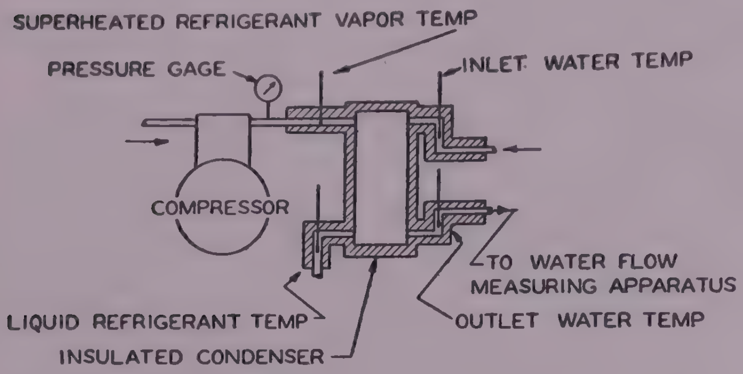


Fig. 4. Condensing Unit with Water Flow Measuring Apparatus.

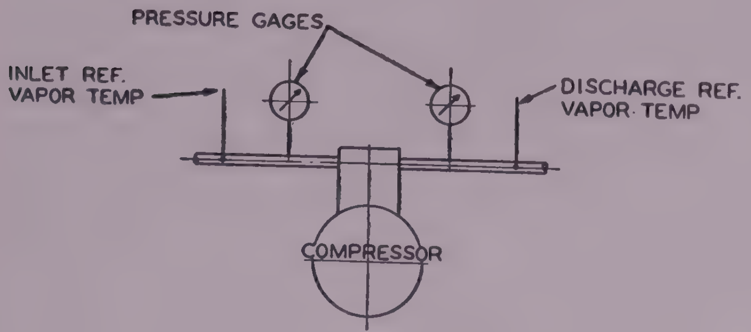


Fig. 5. Calibrated Compressor.

5.27 Remote reading instruments or air sampling tubes shall be used to measure wet and dry bulb temperatures leaving the air treating compartments. Readings shall be taken at not less than six points equally distributed across the area of the respective discharge openings and shall be located not more than six inches in front of the opening. At the point in the sampling tubes where thermometers are inserted, the inside diameter
shall be not less than three inches and the ai
velocity shall be approximately 1000 ft per min bu
not less than 700 ft per min. The air samplin
tube may be brought outside of the calorimete
walls for ease of reading the thermometers bu
must be sealed and insulated to avoid air leakag
and heat leakage. The sampling tube fan and fa
motor should be installed completely within th
calorimeter and its electrical input included in th
load measurement. The fan motor should be lo
cated so that its heat will not affect the temperatur
of air passing into the air conditioner. The fa
shall draw the air over the thermometers and re
turn the air to the same compartment in a manne
that will not affect air temperature measurement
or circulation of air from the air conditioner.

5.28 Interior surfaces of the calorimeter shall be c
nonporous material with all joints sealed agains
air and moisture leakage. Access doors shall b
tightly sealed against air and moisture leakage b
use of gaskets or other suitable means. The floo
of the outdoor compartment may be in the form o
a metal pan to catch any water that may be di
charged by the condenser air. The calorimeter, in
cluding its separating partition, shall be insulate
to prevent heat leakage exceeding 5.0 percent o
the air conditioner capacity. Studding and hig
quality of home insulation may be used.

5.29 Heat leakage shall be determined with all open
ings closed except the equalizing opening describe
in Paragraph 5.24 which shall be closed with
cloth damper normally used to prevent gravity ci
culation of air. Heat leakage from the room com
partment shall be determined by measuring inpu
to the electric heaters required to maintain th
compartment not less than 20 deg F above the sur
rounding ambient temperature The outdoor com
partment shall be at the surrounding ambient tem
perature. Heat leakage from the outdoor compar
ment shall be determined by measuring the hea
removed by the cooling coil while maintaining th
compartment not less than 20 deg F below the sur
rounding ambient temperature, or electric heate
may be temporarily installed and heat leakage de
termined in the same manner as for the room com
partment. During this test the room compartme
shall be at the same temperature as the surroun
ing ambient. If construction of the separating pa
tition between the room and outdoor compartme
is identical with construction of the remainder
the calorimeter, it may be calculated on a propo
tional area basis; if not identical, it may be mea
ured by repeating either of the tests describe
above with the adjoining compartment at a tem
perature equal to that of the compartment unde
test. Heat leakage through the partition then
the difference between the results obtained wit
both compartments at the same temperature an
with the compartments at a minimum differenti
of 20 deg F. During the determination of hea
leakage and during the operation of a test, th
surrounding ambient shall be maintained at
constant temperature plus or minus 2 deg F
determined by not less than one temperature mea
uring instrument located one foot from each sid
and the top of the calorimeter.

5.30 *Condensing Unit with Water Flow Measuring Apparatus (Fig. 4)*

5.31 This apparatus measures the flow rate of r
frigerant circulated to the air conditioner by d

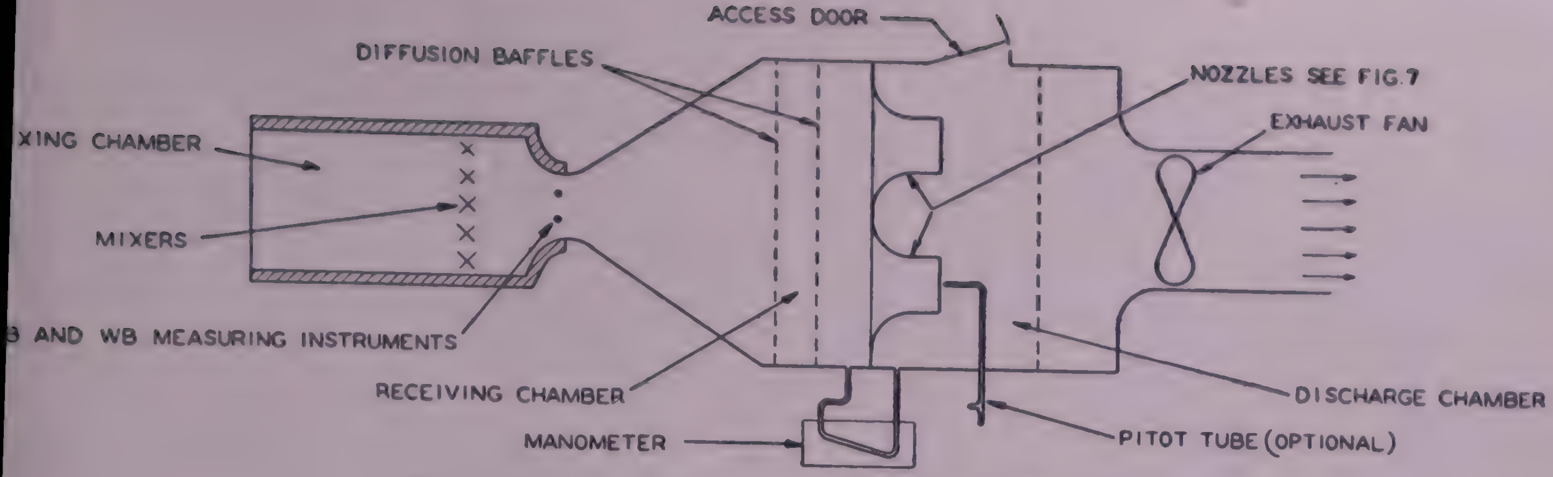


Fig. 6. Air Flow and Temperature Measuring Apparatus.

termining its change in enthalpy and the heat removed on passing through the condenser. This refrigerant flow rate is then used to determine the refrigeration effect within the air conditioner.

5.32 As illustrated in Figure 4, temperature and pressure of the superheated refrigerant vapor are measured as it enters the condenser and the temperature of liquid refrigerant is measured as it leaves the condenser. Refrigerant vapor entering the condenser should be superheated not less than 10 deg F and refrigerant liquid leaving the condenser should be subcooled not less than 5 deg F or means provided to indicate that no vapor is present. Water temperature is measured as it enters and leaves the condenser and the water flow rate shall be determined by apparatus described in Section 5.60. The condenser shall be insulated to limit heat loss to not over 5.0 percent of the total heat removal. This heat loss shall be included in the total heat calculations.

.40 *Calibrated Compressor (Fig. 5)*

5.41 This apparatus determines the refrigerant flow rate by calibration of the compressor under conditions identical to test conditions. The refrigerant flow rate is then used to determine the refrigeration effect within the air conditioner.

5.42 The calibrated compressor should be the one installed in the self-contained air conditioner under test or the test compressor for remote type air conditioners. In either case, it should be calibrated before and after test in accordance with "Standard Methods of Rating and Testing Refrigerant Compressors," ASRE Standard 23-R, and in an ambient temperature corresponding to that within the unit during operation at test conditions. The two calibrations shall not differ by more than 2.0 percent. Calibration tests shall be conducted under conditions applicable to the discharge and suction pressures and temperatures experienced during operation of the air conditioner on test. It is good practice to conduct calibration tests at pressures slightly above and below the operating range and note the rate of variation over this range.

5.43 As shown in Figure 5, temperature and pressure of refrigerant vapor are measured as it enters and leaves the compressor. Superheat of the vapor entering the compressor should be not less than 10 deg F. Superheat of the vapor leaving the compressor indicates uniformity during calibration and test operation. It should differ not more than 10 deg F for the same operating conditions.

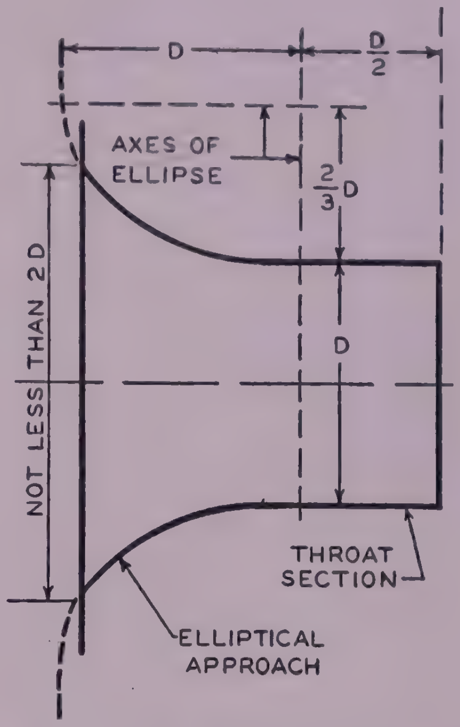


Fig. 7. Flow Measuring Nozzle.

5.50 *Air Flow and Temperature Measuring Apparatus (Fig. 6)*

5.51 Temperature and flow rate of air leaving the conditioner are determined with apparatus shown in Figure 6. The air first passes through a mixing device to eliminate stratification at the point of temperature measurement. Flow rate is determined by measuring the pressure drop across one or more nozzles of the type shown in Figure 7.

5.52 Outlet or outlets of the air conditioner shall be connected to a mixing box at the outlet of which temperature measurements are made by means of sampling apparatus described in Paragraph 5.27 or in openings so restricted as to obtain at least 700 ft per min air velocity over the thermometers. If necessary, the mixing box shall contain deflectors or vanes to mix the air stream to the extent that dry bulb temperatures measured at its outlet shall vary by not more than 1 deg F. The mixing box shall be insulated so that heat leakage shall not exceed 5 percent of the air conditioner capacity. Heat leakage shall be calculated and correction of the air conditioner capacity made accordingly. The ambient temperature used in this calculation shall be determined by not less than two temperature measuring instruments. This mixing box shall be connected to one wall of the receiving

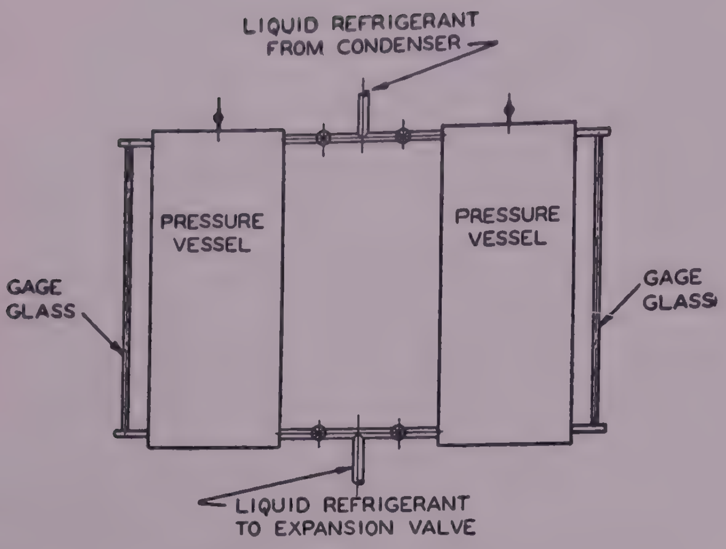

LIQUID REFRIGERANT FROM CONDENSER
PRESSURE VESSEL
PRESSURE VESSEL
GAGE GLASS
GAGE GLASS
LIQUID REFRIGERANT TO EXPANSION VALVE

FIG. 8. CALIBRATED PRESSURE VESSELS FOR VOLATILE LIQUIDS.

chamber. Reference to a suggested design for an air mixing device is given in Paragraph 5.26.

5.53 One or more nozzles constructed in accordance with Figure 7 shall be fitted into one wall of the receiving chamber, Figure 6, discharging into a discharge chamber, and shall be of such a size that the throat velocity is not less than 3000 ft per min. Center-to-center distances between nozzles in use shall not be less than three throat diameters and the distance from the center of any nozzle to any of the four adjacent side walls shall be not less than 1.5 throat diameters. If the nozzles are of different diameters, the distance between axes shall be based upon the average diameter. Size and arrangement of the receiving chamber shall be sufficient to provide uniform approach velocity to the nozzle(s) or shall have suitable diffusion baffles to accomplish this purpose.

5.54 Nozzles installed in accordance with Paragraph 5.53 may be considered to have a negligible correction for approach velocity.

5.55 Size and arrangement of the discharge chamber shall be such that the distance from the center of any nozzle to the adjacent side wall shall not be less than 1.5 throat diameters and not less than five throat diameters to the next obstruction unless suitable diffusion baffles are used. If desired, the discharge chamber may be provided with an access door.

5.56 An exhaust fan shall be connected to the discharge chamber to overcome the resistance of chamber, nozzle(s), and mixing box. It shall be able to provide a zero static pressure at the outlet of the air conditioner.

5.57 To measure the pressure drop across nozzle(s), one or more manometers in parallel shall have one side connected to one or more static pressure connections located flush with the inner wall of the receiving chamber. The other side of the manometer(s) shall be connected in a similar manner to one or more static pressure connections in the wall of the discharge chamber. Static pressure connections shall be located so as not to be affected by air flow. If desired, the velocity head of the air stream leaving the nozzle(s) may be measured by a pitot tube as indicated in Figure 6 but, when more than one nozzle is in use, the pitot tube reading shall be determined for each nozzle. Temperature readings at the nozzle(s) shall be used only for determining air density.

5.60 *Liquid Flow Measuring Apparatus*

5.61 *Liquid Quantity Meter for Nonvolatile Liqui*
shall have an accuracy as specified in Paragrap
4.61. It shall consist of a tank having sufficie
capacity to accumulate the flow for at least tw
minutes and located so that the liquid leaving th
unit under test can be diverted into it. Scale
having an accuracy as specified in Paragraph 4.9
shall be used to weigh the intercepted fluid.

5.62 *Liquid Flow Meter for Nonvolatile Liqui*
shall have an accuracy as specified in Paragrap
4.61.

5.63 *Calibrated Pressure Vessels for Volatile Liquid.*
Figure 8, shall consist of two vertical pressure ve
sels about four feet high connected in parallel by
horizontal header at the top and one at the botton
These headers shall have shut-off valves installe
in them between the entering connection and th
tanks. Each vessel shall be fitted with a graduate
gage glass covering approximately its entire lengt
The charge of refrigerant used in the measurin
vessels shall be new or distilled to avoid addin
oil to the system.

Table 3. Recommended Sizes of Volume Meter
Pressure Vessels for Volatile Liquids

Diam, in.	*Length, ft*	*Range of use, tons*	*Refrigerant*
4	4	0 to 2	Freon (F-12)
4	4	0 " 2	Sulfur dioxide
4	4	0 " 2	Methyl chloride
4	4	2 " 7½	Ammonia
8	4	2 " 10	Freon (F-12)
8	4	2 " 10	Sulfur dioxide
8	4	2 " 10	Methyl chlorid
8	4	7½ " 40	Ammonia
12	4	10 " 25	Freon (F-12)
12	4	10 " 25	Sulfur dioxide
12	4	10 " 25	Methyl chlorid

The vessels and their connections shall be thermall
insulated. Each measuring vessel shall be cal
brated with a liquid of known density, and th
gage glass scales shall be marked in accordanc
with this calibration. Calibrations are recom
mended for various sizes of vessels, as follows:

4-in. diameter, in hundredths (1/100) cu ft
8-in. diameter, in tenths (1/10) cu ft
12-in. diameter, in fourths (1/4) cu ft

5.64 *Liquid Quantity Meter for Volatile Refrigerant*
may be a quantity meter, or a flow meter with a
curacy as specified in Paragraph 4.61. It shall b
connected in the liquid line between the liqui
receiver outlet and the refrigerant control devic
In order that the meter may function properl
under all conditions, additional apparatus shall b
provided as follows:

a. A subcooler ahead of the meter to prevent va
porization of refrigerant in the meter
b. Sight glasses located immediately before an
after the meter to observe that vapor bubbles a
not mixed with the liquid refrigerant
c. A thermometer and well for measuring the ten
perature at which liquid refrigerant leaves th
meter
d. An oil separator located in the compressor di
charge line with provision for returning oil
the compressor crankcase, either manually
automatically

.70 *Steam Condensate Measuring Apparatus (Fig. 9)*

5.71 This apparatus provides a supply of superheated steam and measures the flow rate of condensate from the air conditioner.

5.72 Where steam is available under sufficient pressure, it may be superheated by first passing it through a separator and then expanding through a throttle valve. Where sufficient steam pressure is not available to develop the required 5 deg F of superheat by expansion, superheating may be accomplished by addition of electric heat or other heating means. It is not necessary to measure the amount of heat thus added.

5.73 Steam pressure indicated on the manometer at the coil inlet shall not vary more than ½ in. of mercury above or below the specified pressure. An automatic pressure regulator may be used in combination with the throttling valve to obtain the necessary regulation.

5.74 Thermometers or other temperature measuring means specified in Section 4.10, inserted in suitable wells, shall be used to measure the temperature of the superheated steam entering the coil and the condensate leaving the coil.

5.75 Condensate flow rate shall be measured by a weighing tank or other means described in Section 5.60. Flashing of the condensate shall be prevented by a subcooler of the water interchanger type shown in Figure 9 or of the finned tube type which dissipates heat of the condensate to the surrounding air.

5.76 Visible means such as a gage glass shall be provided to insure that no steam enters the condensate subcooler. The throttling valve shown in Figure 9 shall be adjusted to maintain the condensate level substantially constant during the test.

Section 6. AIR FLOW MEASUREMENT

6.10 *Apparatus*

6.11 Air flow shall be measured with apparatus described in Section 5.10 and illustrated in Figure 6. The principle of this method is to measure the quantity and the temperature change of air entering and leaving an air conditioner.

6.20 *Nozzles*

6.21 When nozzles are constructed in accordance with Figure 7 and installed in accordance with Paragraph 5.53, they may be used without calibration. If the throat diameter is five inches or larger, the coefficient may be assumed to be 0.99. For nozzles smaller than five inches in diameter, or where a more precise value of coefficient is desired, Table 4 may be used or, preferably, the nozzle should be calibrated.

Table 4. Flow Coefficients for Nozzles

Reynolds number, R	*Flow coefficient, Cd*
50,000	0.97
100,000	0.98
150,000	0.98
200,000	0.99
250,000	0.99
300,000	0.99
400,000	0.99
500,000	0.99

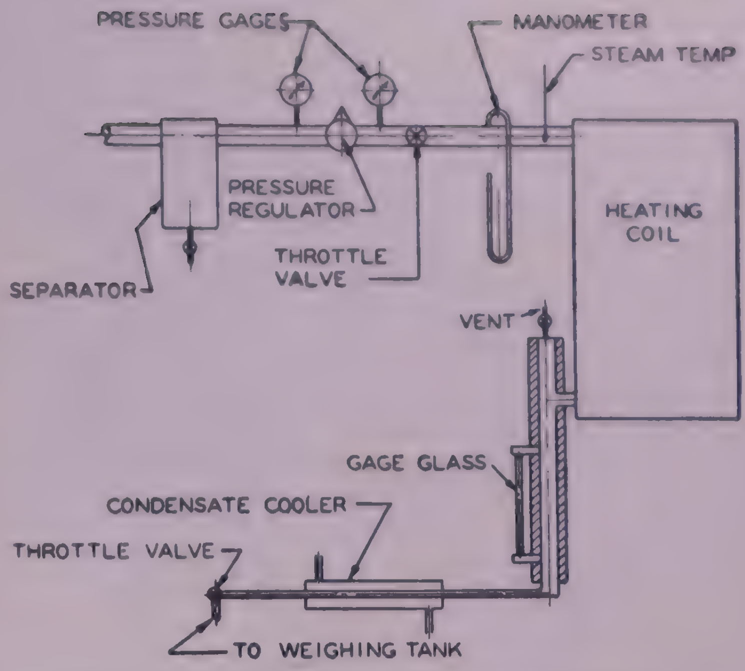


FIG. 9. STEAM CONDENSATE MEASURING APPARATUS.

Reynolds number R for air may be determined from:

$$R = KVD \qquad (1)$$

where K = a factor depending on temperature as given in Table 5
D = throat diameter of nozzle, in.
V = velocity at throat, fpm

Table 5. Factor for Use in Determining Reynolds Number

Temperature, F	*Factor, K*
20	10.1
40	9.3
60	8.7
80	8.1
100	7.5
120	7.1
140	6.7
160	6.3

6.30 *Calculation of Air Flow*

6.31 Data to be recorded:

a. Nozzle throat diameter, in. (D)
b. Nozzle area, sq ft (A)
c. Number of nozzles in use
d. Static pressure difference across nozzle, or velocity pressure if pitot tube is used, in. of water (h)
e. Dry bulb temperature of air at nozzle, F (t_n)
f. Wet bulb temperature of air at nozzle, F (t'_n)
g. Barometric pressure, in. Hg (P)
h. Specific volume of air at nozzle, cu ft per lb of dry air (v_n)
i. Nozzle coefficient (C_d) (see Par. 6.21)

6.32 Air flow, Q_n in cfm, through a single nozzle

$$Q_n = 1096\ C_d A \sqrt{h v'_n} \qquad (2)$$

$$v^1_n = \frac{29.92 v_n}{(1+W)\ P} \qquad (3)$$

where C_d = nozzle coefficient (see Par. 6.21)
A = nozzle area, sq ft
h = velocity pressure at nozzle throat or static pressure difference across nozzle, in. of water

v_n = specific volume of air at dry and wet bulb temperature conditions existing at nozzle but at standard barometric pressure, cu ft per lb of dry air
v'_n = specific volume of air at nozzle, cu ft per lb of air-water vapor mixture
W = specific humidity of air at nozzle, lb of moisture per lb of dry air
P = barometric pressure, in. Hg

For the range of conditions specified in Table 1 and where the barometric pressure deviates from 29.92 in. Hg by not more than one in. Hg, v'_n may, for simplicity, be considered equal to v_n.

6.33 Air flow through multiple nozzles shall be calculated in accordance with paragraph 6.32 except that total flow rate will be the sum of the Q_n for each nozzle used.

6.34 The flow rate of standard air (Q_o) in cfm is:

$$Q_o = \frac{Q_n}{v_n \times 0.075} \qquad (4)$$

where Q_n = measured air flow rate at nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air

6.35 Air flow from room air conditioners (Q_o), in cfm, is:

$$Q_o = \frac{q_t\, v_o}{(H_1 - H_2)\,60} = \frac{.222\, q_t}{H_1 - H_2} \qquad (5)$$

where q_t = net total cooling effect, Btu per hr
v_o = specific volume of standard air, cu ft per lb of dry air
H_1 = enthalpy of entering air, Btu per lb of dry air
H_2 = enthalpy of leaving air, Btu per lb of dry air

Section 7. LIQUID FLOW MEASUREMENT

7.10 *Apparatus*

7.11 Liquid flow shall be measured using apparatus described in Sections 5.30, 5.40, 5.60, and 5.70.

7.20 *Nonvolatile Refrigerant and Steam Condensate Flow Measurements*

7.21 The flow shall be calculated from the following formula:

$$w = (60M)/T \qquad (6)$$

where w = rate of liquid flow, lb per hr
M = weight of liquid collected, lb
T = time to collect liquid, min

or directly from the calibration chart for the flow meter, when that instrument is used.

7.30 *Volatile Refrigerant Flow Measurement in Liquid Phase, Using Calibrated Pressure Vessels*

7.31 Test Procedure. Measurement of the flow of liquid refrigerant is made by timing the interval for the dispensing of a volume of liquid from one of the vessels for a period of not less than two minutes. The diameter of the vessel used should be such that the height of liquid refrigerant dispensed is not less than six (6) inches. This volume is dispensed to the air conditioner being tested by closing the inlet valve on the dispensing vessel and the outlet valve on the collecting vessel. The supply of refrigerant is then furnished continuously to the refrigerant control device. Use new or distilled refrigerant in these pressure vessels at the start to avoid metering oil that might be in circulation i the system. Care should be taken to start timin after a steady flow is established and to finis timing before the level in the vessel supplyin the refrigerant control device falls to a point whic breaks the liquid seal at the outlet. Timing sha be recorded on the vessel in which the liquid leve is falling. Total volume of liquid refrigerant in th two vessels must be constant within 1.0 percen at all times during the test. This total volum shall be determined by simultaneous readings c the liquid levels in the vessels.

7.32 The rate of refrigerant flow shall be deter mined from the following formula:

$$w = \frac{60df\rho}{T} \qquad (7$$

where w = rate of refrigerant flow, lb per hr
d = in. fall in level of liquid in dispens ing vessel
f = cu ft of refrigerant per in. of tan height
ρ = density of refrigerant liquid in dis pensing vessel, lb per cu ft
T = time for level to fall d in., min

7.40 *Volatile Refrigerant Flow Measurement i Liquid Phase, Using Liquid Quantity Mete*

7.41 Rate of refrigerant flow shall be calculated fron the following formula:

$$w = (60B\rho)/T \qquad (8$$

where w = rate of refrigerant flow, lb per hr
B = difference in meter readings, cu f
ρ = density of liquid refrigerant, lb pe cu ft
T = time between meter readings, min

7.50 *Volatile Refrigerant Flow Measurement Using Water Cooled Condenser as Flow Meter*

7.51 Rate of refrigerant flow shall be calculated from the following formula:

$$w = \frac{w_w\,(t_{w2} - t_{w1}) + q_l}{h_g - h_f} \qquad (9)$$

where w = rate of refrigerant flow, lb per hr
w_w = rate of water flow, lb per hr
t_{w2} = temperature of water leaving con denser, F
t_{w1} = temperature of water entering con denser, F
q_l = heat loss from condenser to sur rounding air, Btu per hr
h_g = enthalpy of vapor refrigerant enter ing condenser, Btu per lb
h_f = enthalpy of liquid refrigerant leav ing condenser, Btu per lb

7.60 *Volatile Refrigerant Flow Measurement Using Calibrated Compressor*

7.61 Rate of refrigerant flow shall be calculated from the following formula:

$$w = \frac{q}{h_{g2} - h_{f1}} \qquad (10)$$

where w = rate of refrigerant flow, lb per hr
h_{g2} = enthalpy of gaseous refrigerant leav ing calorimeter, Btu per lb
h_{f1} = enthalpy of liquid refrigerant enter ing calorimeter, Btu per lb
q = capacity of calibrated compresso

as determined from calorimeter, Btu per hr

Section 8. COOLING CAPACITY TESTS

8.00 *Requirement for Rating*

8.01 In order to fulfill the requirements of this standard, two simultaneous methods of checking the ratings shall be used. One method shall determine the cooling capacity on the air or "room" side, the other shall measure the cooling capacity on the refrigerant or "outdoor" side. The two tests shall agree within 6.0 percent.

8.02 Rated capacity of room air conditioners for sensible, latent, and total heat shall be determined from the room side.

8.03 Rated capacity for all air conditioners other than room air conditioners shall be the average of the two capacities specified in Paragraph 8.01. Rated sensible heat capacity shall be determined by multiplying the rated (average) total heat capacity by the ratio of sensible to total heat determined by air side measurements. Rated latent heat capacity shall be determined by multiplying the rated (average) total heat capacity by the ratio of latent heat to total heat determined by air side measurements.

8.04 Standard ratings shall be at a barometric pressure of 29.92 in. Hg. For barometric pressures other than 29.92 in. Hg, dry and wet bulb temperatures shall be maintained as specified in Table 1; the dew point or relative humidity will then differ from Table 1 in accordance with psychrometric data contained in the 1949 Basic Volume of ASRE *Data Books*. For forced circulation air conditioners, no change shall be made in fan speed or system resistance to correct for variations of barometer, but the air quantity shall be corrected to the standard pressure. If desired, rating can be increased by 0.8 percent for each inch of barometer reading below 29.92 in. Hg at which the test was conducted.

Table 6. Data To Be Recorded for Cooling Capacity Tests

		Air Conditioner Type →	*Room air conditioner, self-contained, air or evaporative cooled condenser*	*Room air conditioner, self-contained, water cooler condenser*	*Air conditioner, self-contained, air or evaporative cooled condenser*	*Air conditioner, self-contained, water cooled condenser*	*Room air conditioner, remote type*	*Air conditioner, remote type for installation within conditioned space*	*Air conditioner, remote type for installation outside conditioned space*	*Air conditioner, absorption type*
		Apparatus Prescribed in Paragraph →	*5.01*	*5.02*	*5.03*	*5.04*	*5.05*	*5.06*	*5.06*	*5.07*
Item no.	*Item*	*Unit*	*Record item marked "X"*							
1	Date	—	X	X	X	X	X	X	X	X
2	Observers	—	X	X	X	X	X	X	X	X
3	Barometric pressure	in. Hg	X	X	X	X	X	X	X	X
4	Speed of cooling fan(s) (where such speed is adjustable or variable)	rpm	X	X	X	X	X	X	X	X
5	Power consumption for each motor	watts	X	X	X	X	X	X	X	X
6	Applied voltage for each motor	volts	X	X	X	X	X	X	X	X
7	Temperature of air surrounding air conditioner	deg F	X	X	X	X	X	X	X	X
8	Total power input to air conditioner	watts	X	X	X	X	X	X	X	X
9	Dry bulb temperature of room air entering unit	deg F	X	X	X	X	X	X	X	X
10	Wet bulb temperature of room air entering unit	deg F	X	X	X	X	X	X	X	X
11	Dry bulb temperature of air discharged by unit	deg F	X	X	X	X	X	X	X	X
12	Wet bulb temperature of air discharged by unit	deg F	X	X	X	X	X	X	X	X
13	Dry bulb temperature of an entering condenser	deg F	X		X					
14	Wet bulb temperature of an entering condenser									
15	Total power input to evaporator side heat loads	watts	X	X			X			
16	Water quantity evaporated in evaporator side humidifier	lb/hr	X	X			X			
17	Cooling water flow rate through condenser side heat rejection coil	lb/hr	X							
18	Temperature of water entering condenser side heat rejection coil	deg F	X							
19	Temperature of water leaving condenser side heat rejection coil	deg F	X							
20	Amount of moisture collected by air treating coil in outdoor comp.	lb/hr	X							
21	Flow rate of cooling water	lb/hr	X	X		X				X
22	Temperature of cooling water entering air conditioner	deg F		X		X				X
23	Temperature of cooling water leaving air conditioner	deg F		X		X				X
24	Pressure difference between cooling water inlet and outlet of water cooled air conditioner with water valve set to shut off at 90 F condensing temp	psi		X		X				X
25	Same as 24 without water regulating valve	psi		X		X				X
26	Pressure of water entering unit	psig		X	X	X				X
27	Static pressure difference between air inlet and air discharge	in. H_2O			X	X		X	X	X
28	Dry bulb temperature of room air at inlet to enclosure	deg F			X	X		X		
29	Wet bulb temperature of room air at inlet to enclosure	deg F			X	X		X		
30	Refrigerant flow rate	lb/hr					X	X	X	
31	Pressure of refrigerant entering unit	psig					X	X	X	
32	Pressure of refrigerant leaving unit	psig					X	X	X	
33	Temperature of refrigerant entering unit	deg F					X	X	X	
34	Temperature of refrigerant leaving unit	deg F					X	X	X	
35	Net heat input to absorption unit	Btu/hr								X

8.05 Test conditions selected from Table 1 shall be maintained until equilibrium has been reached, but for not less than one hour, before recording data for the rating test. The test shall then be run for one hour recording data every ten minutes, giving seven sets of readings.

8.06 Table 6 lists required data. Items marked with an "X" under each of the equipment classifications shall be recorded. The table shows general information required but is not intended to limit the data to be obtained. The table also indicates the paragraph numbers in Section 5 which specify the apparatus to be used.

8.10 *Room Air Conditioners, Self-Contained, with Air or Evaporatively Cooled Condensers*

8.11 Room air conditioners, self-contained, with air or evaporatively cooled condensers of the type installed in or in front of a window, shall be tested using the apparatus designated in Paragraph 5.01.

8.12 Net total room cooling effect of self-contained room air conditioners tested using the calorimeter shown in Figure 3 shall be the capacity determined on the room air side.

8.13 Net total room cooling effect on the room air side, q_{tr}, in Btu per hr, is calculated as follows:

$$q_{tr} = 3.41\,\Sigma E_r + q_{lr} + (h_{w1} - h_{w2})\,w_r \qquad (11)$$

where ΣE_r = sum of all power input to room air side of calorimeter, watts

q_{lr} = heat leakage into or out of room air compartment as calculated from calibrating test, Btu per hr
Calculation for q_{lr} should include outside walls of room compartment and separating partition considered separately. This value is positive for heat leakage into this compartment, and negative for heat leakage out of the compartment.

h_{w1} = enthalpy of moisture supplied to the humidifier or steam. If no water is added during the test, h_{w1} is taken at the temperature of the stored water. Btu per lb

h_{w2} = enthalpy of moisture leaving the room side of the calorimeter. May be assumed to be at the wet bulb temperature of the air leaving the air conditioner. Btu per lb

w_r = amount of water evaporated in room air circuit to maintain humidity required, lb of water per hr

8.14 Net total room cooling effect on the condenser side, q_{ta}, in Btu per hr, is calculated as follows:

$$q_{ta} = q_c - 3.41\,\Sigma E_a - 3.41\,E + q_{la} + (t_w - 32)\,w_m \qquad (12)$$

where q_c = heat removed by cooling coil in outdoor compartment of calorimeter, Btu per hr

ΣE_a = sum of all power input to any calorimeter equipment, such as circulating fans, etc., in condenser air side of calorimeter, watts

E = total power input to air conditioner, watts

q_{la} = heat leakage out of or into condenser air compartment as calculated from calibrating test, Btu per hr

t_w = temperature of condensate leaving the calorimeter

w_m = moisture removed by air treating coil in calorimeter outdoor compartment, lb per hr
Calculation for q_{la} should include outside walls of condenser air compartment and separating partition considered separately. This value is positive for heat leakage out of compartment, and negative for heat leakage into compartment.

8.15 If the air—or evaporatively—cooled condenser is remote from the air conditioner, the condenser should be separately rated and the air conditioner should be tested with its condensing temperature and liquid temperature fixed as shown in Table 1. Then the net total room cooling effect on the condenser side, q_{tf}, in Btu per hr, is calculated as follows:

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E \qquad (13)$$

where w_f = weight flow of refrigerant as determined in Section 7, lb per hr

h_{g2} = enthalpy of refrigerant vapor leaving air conditioner, Btu per lb of refrigerant

h_{f1} = enthalpy of refrigerant liquid entering air conditioner, Btu per lb of refrigerant

E = total power input to air conditioner, watts

8.16 Net total room dehumidifying effect, q_d, in Btu per hr, of self-contained air conditioners tested using calorimeter shown in Figure 3 is calculated as follows:

$$q_d = 1060 w_r \qquad (14)$$

where w_r = amount of water evaporated in room air circuit to maintain humidity required, lb of water per hr

8.17 Net total sensible cooling effect, q_s, in Btu per hr, shall be determined by subtracting q_d, the net room dehumidifying effect, from q_t, the average net total room cooling effect, or

$$q_s = q_t - q_d \qquad (15)$$

$$\text{where } q_t = (q_{tr} + q_{ta})/2 \qquad (16)$$

8.20 *Room Air Conditioners, Self-Contained, with Water Cooled Condensers*

8.21 Room air conditioners, self-contained, with water cooled condensers shall be tested using apparatus designated in Paragraph 5.02.

8.22 Net total room cooling effect of water cooled room air conditioners tested using calorimeter shown in Figure 3 shall be the average of the capacities determined on the room air side and on the condenser side.

8.23 Net total room cooling effect on the room air side, q_{tr}, should be calculated from Equation 11.

8.24 Net total room cooling effect on the condenser side, q_{tw}, in Btu per hr, is calculated as follows:

$$q_{tw} = w_w\,(t_{w2} - t_{w1}) - 3.41\,E \qquad (17a)$$

where w_w = water flow through condenser, lb per hr

t_{w2} = temperature of condenser cooling water leaving unit, F

t_{w1} = temperature of condenser cooling water entering unit, F

E = total power input to air conditioner, watts

For room air conditioners which take ventilation air from within the air conditioned space and discharge it outside the enclosure, the condenser side total room cooling effect, q_{tw}, in Btu per hr, will be as follows:

$$q_{tw} = w_w\,(t_{w2} - t_{w1}) - 3.41\,E + w_v\,(H_{v2} - H_{r1}) \qquad (17b)$$

where w_w = water flow through condenser, lb per hr
t_{w2} = temperature of condenser cooling water leaving air conditioner, F
t_{w1} = temperature of condenser cooling water entering air conditioner, F
E = total power input to air conditioner, watts
w_v = ventilation air discharge quantity, lb per hr
H_{v2} = enthalpy of air leaving ventilation discharge of air conditioner, Btu per lb of dry air
H_{r1} = enthalpy of air entering enclosure, Btu per lb of dry air

8.25 Net total room dehumidifying effect, q_d, of self-contained air conditioners tested using calorimeter shown in Figure 3 is calculated from Equation 14.

8.26 Net total sensible cooling effect, q_s, is determined as shown in Equation 15.

.30 *Air Conditioners, Self-Contained, with Air or Evaporatively Cooled Condensers*

8.31 Air conditioners, self-contained, with air or evaporatively cooled condensers shall use the apparatus designated in Paragraph 5.03. If the air conditioner is intended for use within or partly within the conditioned space, equipment designated by either Paragraph 5.03a or 5.03b may be used but if intended for use outside the conditioned space, equipment described in Paragraph 5.03b must be used. Where the calibrated room calorimeter described in Section 5.20 is used, the test shall be conducted and the calculations made according to Section 8.10. Where the psychrometric calorimeter described in Section 5.10 is used, the following calculations shall be followed.

8.32 Net total room cooling effect, q_{tr}, of self-contained air conditioners with air or evaporatively cooled condensers shall be the average of capacities determined on the room air side and the condenser side. When no ventilation air is circulated through the unit, or where the ventilation air is discharged from the unit to within the test enclosure, or where the ventilation air is discharged from the unit through the room air discharge of the unit, the net total room cooling effect, q_{tr}, measured on the air side is determined from Equation 18a if the unit is intended for use within the space to be conditioned, and from Equation 18b if intended for use outside the space to be conditioned

$$q_{tr} = 60 \frac{Q_n}{v_n} (H_{m1} - H_{r2}) \qquad (18a)$$

$$q_{tr} = 60 \frac{Q_n}{v_n} (H_{r1} - H_{r2}) \qquad (18b)$$

where Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
H_{m1} = enthalpy of air entering enclosure, Btu per lb of dry air
H_{r2} = enthalpy of air leaving room air discharge of air conditioner, Btu per lb of dry air
H_{r1} = enthalpy of air entering air conditioner, Btu per lb of dry air

8.33 Net air side room dehumidifying effect, q_{dr}, in Btu per hr, shall be calculated from Equation 19a if intended for use inside the conditioned space and Equation 19b if intended for use outside the conditioned space:

$$q_{dr} = 1060 \times 60 \frac{Q_n}{v_n} (W_{m1} - W_{r2}) \qquad (19a)$$

$$q_{dr} = 1060 \times 60 \frac{Q_n}{v_n} (W_{r1} - W_{r2}) \qquad (19b)$$

where Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
W_{m1} = specific humidity of air entering calorimeter from room, lb of water per lb of dry air
W_{r2} = specific humidity of air leaving room air discharge of unit, lb of water per lb of dry air
W_{r1} = moisture content of air entering unit, lb of water per lb of dry air

8.34 Net room sensible cooling effect, q_{sr}, in Btu per hr as measured on the room air side, shall be calculated from Equation 20a if intended for use inside the conditioned space, and Equation 20b if intended for use outside the conditioned space:

$$q_{sr} = 60 \frac{Q_n}{v_n} (t_{m1} - t_{r2}) \times .24 \qquad (20a)$$

$$q_{sr} = 60 \frac{Q_n}{v_n} (t_{r1} - t_{r2}) \times .24 \qquad (20b)$$

where Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
t_{m1} = dry bulb temperature of air entering enclosure
t_{r2} = dry bulb temperature of air leaving discharge of air conditioner
t_{r1} = dry bulb temperature of air entering air conditioner

8.35 Net total room cooling effect in Btu per hr as measured on the condenser side, q_{ta}, shall be calculated from Equation 21a if the unit is intended for installation inside the conditioned space, and Equation 21b if intended for use outside the conditioned space:

$$q_{ta} = w_a (H_{a2} - H_{a1}) - 3.41\,E \qquad (21a)$$

$$q_{ta} = w_a (H_{a2} - H_{a1}) - 3.41\,E + 60 \frac{Q_n}{v_n} (H_{r1} - H_{m1}) \qquad (21b)$$

where w_a = condenser air flow, lb per hr
H_{a2} = enthalpy of air leaving outside air discharge of air conditioner, Btu per lb of dry air
H_{a1} = enthalpy of air entering outside air intake of air conditioner, Btu per lb of dry air
E = total power input to air conditioner, watts
Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
H_{r1} = enthalpy of air entering unit, Btu per lb of dry air
H_{m1} = enthalpy of air entering enclosure, Btu per lb of dry air

For self-contained air conditioners tested using enclosure shown in Figure 1 when ventilation air is taken from within the test enclosure and is discharged outside the enclosure other than through the room air discharge of the air conditioner, the condenser side total room cooling effect, q_{ta}, will be determined by Equation 21c for units intended for use inside the conditioned space, and Equation 21d for units intended for installation outside the conditioned space:

$$q_{ta} = w_a - (H_{a2} - H_{a1}) - 3.41\,E + w_v\,(H_{v2} - H_{r1}) \qquad (21c)$$

$$q_{ta} = w_a - (H_{a2} - H_{a1}) - 3.41\,E + w_v\,(H_{v2} - H_{r1}) + 60\,\frac{Q_n}{v_n}\,(H_{r1} - H_{m1}) \qquad (21d)$$

where w_a = condenser air flow, lb per hr

H_{a2} = enthalpy of air leaving outside air discharge of air conditioner, Btu per lb of dry air

H_{a1} = enthalpy of air entering outside intake of air conditioner, Btu per lb of dry air

E = total power input to air conditioner, watts

w_v = ventilation air discharge quantity, lb per hr

H_{v2} = enthalpy of air leaving ventilation discharge of air conditioner, Btu per lb dry air

H_{r1} = enthalpy of air entering enclosure, Btu per lb of dry air

Q_n = air flow through nozzle, cfm

v_n = specific volume of air at nozzle, cu ft per lb of dry air

H_{r1} = enthalpy of air entering unit, Btu per lb of dry air

H_{m1} = enthalpy of air entering enclosure, Btu per lb of dry air

8.36 If the condenser is remote from the air conditioner, it shall be separately rated and the air conditioner shall be tested with its condensing temperature and liquid temperature fixed as shown in Table 1. Then the net total room cooling effect, in Btu per hr as measured on the refrigerant side, q_{tf}, shall be calculated from Equation 22a if the unit is intended for installation inside the conditioned space, and Equation 22b if intended for use outside the conditioned space:

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E \qquad (22a)$$

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E + 60\,\frac{Q_n}{v_n}\,(H_{r1} - H_{m1}) \qquad (22b)$$

where w_f = weight flow of refrigerant as determined in Section 7, lb per hr

h_{g2} = enthalpy of refrigerant vapor leaving air conditioner, Btu per lb of refrigerant

h_{f1} = enthalpy of refrigerant liquid entering air conditioner, Btu per lb of refrigerant

E = total power input to air conditioner, watts

Q_n = air flow through nozzle, cfm

v_n = specific volume of air at nozzle, cu ft per lb of dry air

H_{r1} = enthalpy of air entering unit, Btu per lb of dry air

H_{m1} = enthalpy of air entering enclosure, Btu per lb of dry air

For self-contained air conditioners tested using enclosure shown in Figure 1 when ventilation air is taken from within the test enclosure and is discharged outside the enclosure other than through the room air discharge of the conditioner, and when the condenser is remote from the air conditioner, the condenser shall be rated separately and the refrigerant side total room cooling effect, q_{tf}, will be determined by Equation 22c for units intended for use inside the conditioned space, and Equation 22d for units intended for installation outside the conditioned space:

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E + w_v\,(H_{v2} - H_{r1}) \qquad (22c)$$

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E + w_v\,(H_{v2} - H_{r1}) + 60\,\frac{Q_n}{v_n}\,(H_{r1} - H_{m1}) \qquad (22d)$$

where w_f = weight flow of refrigerant as determined in Section 7, lb per hr

h_{g2} = enthalpy of refrigerant vapor leaving air conditioner, Btu per lb of refrigerant

h_{f1} = enthalpy of refrigerant liquid entering the air conditioner, Btu per lb of refrigerant

E = total power input to the air conditioner, watts

w_v = ventilation air discharge quantity, lb per hr

H_{v2} = enthalpy of air leaving ventilation discharge of air conditioner, Btu per lb of dry air

H_{r1} = enthalpy of air entering unit, Btu per lb of dry air

Q_n = air flow through nozzle, cfm

v_n = specific volume of air at nozzle, cu ft per lb of dry air

H_{m1} = enthalpy of air entering enclosure, Btu per lb of dry air

8.40 *Air Conditioners, Self-Contained, with Water Cooled Condensers*

8.41 Air conditioners, self-contained, with water cooled condensers, when intended for use inside or outside of the conditioned space, shall use the apparatus designated in Paragraph 5.04 except that air conditioners intended for use inside the conditioned space may use the calibrated room calorimeter described in Section 5.20, in which case the test shall be conducted and the calculations made according to Section 8.10. Where the calibrated room calorimeter is not used, the following procedure shall be followed. The psychrometric calorimeter shall be used with an enclosure for ratings to be used both when the unit is intended for installation within a conditioned space, and when the unit is intended for installation outside of the conditioned space. "Inlet" temperatures from Table 1 shall be those measured at the entrance to the enclosure for units intended for installation within the conditioned space and at the entrance to the unit when the unit is intended for installation only remote from the conditioned space. Note importance of accurate dry and wet bulb temperature readings as indicated in Paragraphs 3.13 and 5.16.

8.42 Net total cooling effect, q_{tr}, of self-contained air conditioners with water cooled condensers shall be the average of the room air and condenser side capacities. When no ventilation air is circulated through the unit, or where the ventilation air is discharged from the unit to within the test enclosure, or where the ventilation air is discharged from the unit through the room air discharge of the unit, the net total room cooling effect in Btu per hour, measured on the air side, is calculated from Equation 18a if intended for use inside the conditioned space, and Equation 18b if intended for use outside the conditioned space.

8.43 Net air side room dehumidifying effect in Btu per hr, q_{dr}, shall be calculated from Equation 19a if intended for use inside the conditioned space, and Equation 19b if intended for use outside the conditioned space.

8.44 Net room sensible cooling effect in Btu per hr as measured on the room air side, q_{sr}, shall be calculated from Equation 20a if intended for use inside the conditioned space, and Equation 20b if intended for use outside the conditioned space.

8.45 Net total room cooling effect in Btu per hr measured on the condenser side, q_{tw}, shall be calculated from Equation 17a if intended for use inside the conditioned space, and Equation 23 if intended for use outside the conditioned space:

$$q_{tw} = w_w\,(t_{w2} - t_{w1}) - 3.41\,E + w_v\,(H_{v2} - H_{r1}) + 60\,\frac{Q_n}{v_n}\,(H_{r1} - H_{m1}) \qquad (23)$$

where w_w = water flow through condenser, lb per hr
t_{w2} = temperature of condenser cooling water leaving air conditioner, F
t_{w1} = temperature of condenser cooling water entering air conditioner, F
E = total power input to unit, watts
w_v = ventilation air discharge quantity, lb per hr
H_{v2} = enthalpy of air leaving ventilation air discharge of air conditioner, Btu per lb of dry air
H_{r1} = enthalpy of air entering unit, Btu per lb of dry air
Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
H_{m1} = enthalpy of air entering enclosure, Btu per lb of dry air

8.50 *Room Air Conditioners, Remote Type (for Use inside Conditioned Space)*

8.51 Room air conditioners, remote type, shall be tested using apparatus designated in Paragraph 5.05.

8.52 Net total room cooling effect, q_t, of remote type room air conditioners shall be the average of capacities determined on the room air side and on the refrigerant side.

8.53 Net total room cooling effect on the room air side, q_{tr}, shall be calculated from Equation 11.

8.54 Net total room dehumidifying effect, q_d, measured on the air side is calculated from Equation 14.

8.55 Net total room cooling effect measured on the refrigerant side, q_{tf}, when a volatile refrigerant is used, is calculated as follows:

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E \qquad (24)$$

where w_f = weight flow of refrigerant as determined in Section 7, lb per hr
h_{g2} = enthalpy of refrigerant vapor leaving evaporator of air conditioner, Btu per lb of refrigerant
h_{f1} = enthalpy of refrigerant liquid entering evaporator of air conditioner, Btu per lb of refrigerant
E = total power input to air conditioner, watts

8.56 Net refrigerant side total room cooling effect measured on the refrigerant side, q_{to}, when a nonvolatile refrigerant is used, is calculated as follows:

$$q_{to} = w_c c\,(t_{c2} - t_{c1}) - 3.41\,E \qquad (25)$$

where w_c = weight flow of refrigerant, lb per hr
c = specific heat of refrigerant, Btu per lb per F
t_{c2} = temperature of refrigerant leaving air conditioner, F
t_{c1} = temperature of refrigerant entering air conditioner, F
E = total power input to air conditioner, watts

8.60 *Air Conditioners, Remote Type, Intended for Use Within or Partly within Conditioned Space*

8.61 These air conditioners shall be tested using apparatus designated in Paragraph 5.06. Where the calibrated room calorimeter described in Section 5.20 is used, the test shall be conducted and the calculations made according to Section 8.10. Where the calibrated room calorimeter is not used, the following calculations shall be followed:

8.62 Net total cooling effect shall be the average of the cooling effects determined on the air side and on the refrigerant side.

8.63 Net total cooling effect on the air side, q_{tr}, is calculated from Equation 18a.

8.64 Net total dehumidifying effect, measured on the air side, q_{dr}, is calculated from Equation 19a.

8.65 Net total cooling effect measured on the refrigerant side, q_{tf}, when a volatile refrigerant is used, is calculated from Equation 24.

8.66 Net total cooling effect measured on the refrigerant side, q_{tc}, when a nonvolatile refrigerant is used, is calculated from Equation 25.

8.70 *Air Conditioners, Remote Type, Intended for Use outside Conditioned Space*

8.71 These air conditioners shall be tested using apparatus designated in Paragraph 5.06. Where the calibrated room calorimeter described in Section 5.20 is used, the test shall be conducted and the calculations made according to Section 8.10. Where the calibrated room calorimeter is not used, the following calculations shall be followed:

8.72 Net total cooling effect shall be the average of the cooling effects determined on the air side and on the refrigerant side.

8.73 Net total cooling effect on the air side, q_{tr}, is calculated from Equation 18b.

8.74 Net total dehumidifying effect, measured on the air side, q_{dr}, is calculated from Equation 19b.

8.75 Net total cooling effect measured on the refrigerant side, q_{tf}, when a volatile refrigerant is used, is calculated from:

$$q_{tf} = w_f\,(h_{g2} - h_{f1}) - 3.41\,E_r \qquad (26)$$

where w_f = weight flow of refrigerant as determined in Section 7, lb per hr
h_{g2} = enthalpy of refrigerant vapor leaving evaporator of air conditioner, Btu per lb of refrigerant
h_{f1} = enthalpy of refrigerant liquid entering evaporator of air conditioner, Btu per lb of refrigerant
E_r = that portion of electric power input to air conditioner which enters conditioned air stream, watts

8.76 Net total cooling effect measured on the refrigerant side, q_{tc}, in Btu per hr, when a nonvolatile refrigerant is used, is calculated as follows:

$$q_{tc} = w_c c\,(t_{c2} - t_{c1}) - 3.41\,E_r \qquad (27)$$

where w_c = weight flow of refrigerant, lb per hr
c = specific heat of refrigerant, Btu per lb per F
t_{c2} = temperature of refrigerant leaving air conditioner, F
t_{c1} = temperature of refrigerant entering air conditioner, F
E_r = that portion of electric power input to air conditioner which enters conditioned air stream, watts

8.80 *Self-Contained Air Conditioners, Absorption Type*

8.81 These air conditioners shall be tested using apparatus designated in Paragraph 5.07. The cooling effect of absorption-type air conditioners suitable for use with duct work shall be obtained using the calorimeter shown in Figure 2 and shall be the average of air and water side capacities. Then, the net total air side cooling capacity, q_{tr}, in Btu per hr shall be calculated as follows:

$$q_{tr} = 60\,\frac{Q_n}{v_n}\,(H_{r1} - H_{r2}) \qquad (28)$$

where Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
H_{r1} = enthalpy of air entering the air conditioner, Btu per lb of dry air
H_{r2} = enthalpy of air leaving the room air discharge of the air conditioner, Btu per lb of dry air

8.82 Net room dehumidifying effect of absorption air conditioners, q_{dr}, in Btu per hr, shall be calculated from test data as follows:

$$q_{dr} = 1060 \times 60\,\frac{Q_n}{v_n}\,(W_{r1} - W_{r2}) \qquad (29)$$

where Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
W_{r1} = moisture content of air to be conditioned, lb of water per lb of dry air
W_{r2} = moisture content of air leaving room air discharge of unit, lb of water per lb of dry air

8.83 Net room sensible cooling effect in Btu per hr as measured on the room air side, q_{sr}, in Btu per hr, shall be calculated as follows:

$$q_{sr} = 60\,\frac{Q_n}{v_n}\,(t_{r1} - t_{r2}) \times .24 \qquad (30)$$

where t_{r1} = temperature of air entering unit, F
t_{r2} = temperature of air leaving room air discharge of unit, F

8.84 Net total room cooling effect on the cooling water side, q_{tw}, in Btu per hr, is calculated as follows:

$$q_{tw} = w_w\,(t_{w2} - t_{w1}) - q_i - 3.41\,E_r \qquad (31)$$

where w_w = water flow through unit, lb per hr
t_{w2} = temperature of cooling water leaving unit (condenser outlet), F
t_{w1} = temperature of cooling water entering unit (inlet to absorber), F
q_i = net heat input to absorption unit, Btu per hr
E_r = that portion of electric power input to air conditioner which enters conditioned air stream, watts

Section 9. HEATING CAPACITY TESTS

9.00 *Requirement for Rating*

9.01 This section describes two basic methods of measuring heating capacity. One is the determination of capacity on the air or "room" side, the other a determination from the heating medium side. It is desirable, where practicable, to have simultaneous tests and to establish the rated capacity as the average from the two methods which should agree within 6.0 percent. In some cases, such as with direct fired heating equipment, it may be impractical to obtain ratings from the heating medium side, in which case the rating from the air or room side should be used.

9.02 Air side measurements of capacity should be made on all air conditioners which have both cooling and heating functions for the reason that the air flow measurements required are necessary for the cooling rating and can be used with a little additional data to obtain heating ratings. Heating air conditioners which do not have cooling functions may be rated only by heating medium measurements in order to minimize the apparatus required. The apparatus required for air side measurements shall be the calibrated room calorimeter prescribed in Section 5.20 and the psychrometric calorimeter prescribed in Section 5.10. The type selected will depend on the type of air conditioner used as specified below.

9.03 Heating medium measurements of capacity should be made using liquid flow measuring apparatus prescribed in Section 5.60 and steam condensate measuring apparatus prescribed in Section 5.70, except where electricity is the heat source or the air conditioner is direct fired. In the case of air conditioners which are direct fired, it will be acceptable to use test apparatus and methods which are in common use such as the American Gas Association test methods for gas fired equipment and the test methods of the Bureau of Standards for oil fired equipment.

9.04 Standard ratings shall be made at a barometric pressure of 29.92 in. Hg. For barometic pressures other than 29.92 in. Hg the dry bulb temperatures shall be maintained as specified in Table 1. For forced circulation air conditioners no change shall be made in fan speed or system resistance to correct for variations of barometer, but the air quantity shall be corrected to the standard pressure. If desired, the heat transfer rate can be corrected by increasing the test results 2.0 percent for each inch of barometer reading below 29.92 in. Hg at which the test was conducted.

.05 Test conditions selected from Table 1 shall be maintained until equilibrium has been reached, but for not less than one hour, before recording data for the rating test. The test shall then be run at least for one hour recording data every ten minutes, giving at least seven sets of readings.

.06 Table 7 lists required data. Items marked with an "X" under each of the equipment classifications shall be recorded. The table shows the general information required but is not intended to limit the data to be obtained.

Table 7. Data To Be Recorded For Heating Capacity Tests

	Air Conditioner Type		*Steam heating*	*Hot water heating*	*Direct fired*	*Heat pump*	*Electric heater*
tem no.	*Item*	*Unit*	*Record items marked "X"*				
	Date	—	X	X	X	X	X
	Observers	—	X	X	X	X	X
	Barometric pressure	in. Hg			X	X	
	Fan speed	rpm	X	X	X	X	
	Power input to fan motor	watts	X	X	X	X	X
	Temp. of air surrounding air conditioner	deg F	X	X	X	X	X
	Steam inlet pressure	psia	X				
	Steam outlet pressure	psia	X				
	Steam inlet temperature	deg F	X				
	Steam condensate outlet temperature	deg F	X				
	Weight flow of steam condensate	lb/hr	X				
	Water inlet temperature	deg F		X			
	Water outlet temperature	deg F		X			
	Weight of water flow	lb/hr		X			
	Air flow	cfm			X	X	
	Air inlet temperature	deg F			X	X	
	Air outlet temperature	deg F			X	X	
	Power input to comp. motor	watts				X	
	Power input to heaters	watts					X

.10 *Room Air Conditioners*

9.11 Room air conditioners, whether self-contained or remote type, of the type installed in or in front of a window shall be tested using the apparatus designated in Section 5.20.

9.12 Net total room heating effect on the air or room side, q_{hr}, in Btu per hr, is calculated as follows:

$$q_{hr} = q_c - 3.41\,\Sigma E_r + q_{lr} \qquad (32)$$

where q_c = heat removed by cooling coil in room side compartment of calorimeter, Btu per hr
ΣE_r = sum of all power input to any calorimeter equipment, such as circulating fans, etc., in room side compartment of calorimeter, watts
q_{lr} = heat leakage out of or into room side compartment as calculated from calibrating test, Btu per hr

Calculation for q_{lr} should include walls of room side compartment and separating partition considered separately. This value is positive for heat leakage out of the compartment and negative for heat leakage into the compartment.

9.13 Net total room heating effect, on the heating medium side, is calculated as specified in Section 9.40 for steam, 9.50 for hot water, 9.60 for electricity, and 9.70 for direct fired.

9.14 In the case of the heat pump, q_{hw}, in Btu per hr, the net total room heating effect on the heating medium side is determined by the heat extraction in the evaporator plus the heat equivalent of the electric energy input. The evaporator would be located in the outside compartment of the calibrated room calorimeter and its heat extraction would be offset by an electric heater and humidifier. Consequently:

$$q_{hw} = 3.41\,\Sigma E_o + q_{lr} + 3.41\,E \qquad (33)$$

where ΣE_o = sum of all power input to outside compartment of calorimeter, watts
q_{lr} = heat leakage out of or into room air compartment as calculated from calibrating test, Btu per hr
E = total power input to air conditioner, watts

Calculation for q_{lr} should include outside walls of room compartment and separating partition considerated separately. This value is positive for heat leakage into this compartment and negative for heat leakage out of the compartment.

9.20 *Air Conditioners for Use within Air Conditioned Room*

9.21 Air conditioners for use within the air conditioned room shall be tested using the psychrometric calorimeter as designated in Section 5.10. Test conditions shall be selected from Table 1 and maintained outside the enclosure.

9.22 Net total room heating effect on the room air side, q_{hr} in Btu per hr, is calculated as follows:

$$q_{hr} = 60\,\frac{Q_n}{v_n}\,(H_{r2} - H_{r1}) \qquad (34)$$

where Q_n = air flow through nozzle, cfm
v_n = specific volume of air at nozzle, cu ft per lb of dry air
H_{r1} = enthalpy of air entering enclosure, Btu per lb of dry air
H_{r2} = enthalpy of air leaving room air discharge of air conditioner, Btu per lb of dry air

9.23 Net total room heating effect on the heating medium side is calculated as specified in Section 9.40 for steam, 9.50 for hot water, 9.60 for electricity, and 9.70 for direct fired. In the case of the heat pump, the net total room heating effect, q_{hw}, in Btu per hr is:

$$q_{hw} = w_f\ (h_{g2} - h_{f1}) + 3.41\ E \qquad (35)$$

where w_f = weight flow of refrigerant as determined in Section 7, lb per hr

h_{g2} = enthalpy of refrigerant vapor leaving evaporator of air conditioner, Btu per lb of refrigerant

h_{f1} = enthalpy of refrigerant liquid entering evaporator of air conditioner, Btu per lb of refrigerant

E = total power input to air conditioner, watts

9.30 *Air Conditioners for Use outside of Conditioned Space*

9.31 Air conditioners for use outside the conditioned space shall be tested using the psychrometric calorimeter as designated in Section 5.10. Test conditions shall be selected from Table 1 and maintained within the enclosure.

9.32 Net total room heating effect on the room air side, q_{hr}, in Btu per hr, is calculated as follows:

$$q_{hr} = 60 \frac{Q_n}{v_n} (H_{r2} - H_{r1}) \qquad (36)$$

where Q_n = air flow through nozzle, cfm

v_n = specific volume of air at nozzle, cu ft per lb of dry air

H_{r1} = enthalpy of air entering air conditioner, Btu per lb of dry air

H_{r2} = enthalpy of air leaving room air discharge of air conditioner, Btu per lb of dry air

9.33 Net total room heating effect on the heating medium side, q_{hw}, is calculated as specified in Section 9.40 for steam, 9.50 for hot water, 9.60 for electricity, and 9.70 for direct fired. In the case of the heat pump, the net total room heating effect, q_{hw}, is calculated from Equation 35.

9.40 *Air Conditioner Where Steam Is Used As Heating Medium*

9.41 For closed steam systems, the weight of steam condensate method consists of weighing the condensate formed in a measured time with apparatus as shown in Section 5. Figure 9 shall be used.

9.42 For atmospheric steam systems, where the steam source is an integral part of a self-contained air conditioner and where air flow is at a rated volume, the condensate shall be collected in a suitable device for weighing. During the limited time condensate is being collected, no make-up water shall be allowed to enter the steam generator.

9.43 Steam shall be supplied from a source sufficient to prevent changes in line pressure from conditions specified in Section 3, Tables 1 and 2.

9.44 The pet cock at the heater outlet shall be opened sufficiently to relieve the system of air.

9.45 Heating capacity, q, in Btu per hr, is as follows:

$$q = w_s\ h_{fg} + 3.41\ E \qquad (37)$$

where w_s = steam condensate, lb per hr

h_{fg} = latent heat of vaporization for steam, Btu per lb

E = electric power input to fan motor, watts

9.50 *Air Conditioner Using Water As Heating Medium*

9.51 The water temperature drop method consists of measuring the heat extracted from the water as determined from measurement of the rate of water flow and the temperature drop of the water through the unit.

9.52 Hot water heating units shall be provided with sufficient means for supplying water at the conditions specified in Section 3, Tables 1 and 2.

9.53 Heating capacity, q, in Btu per hr, is as follows:

$$q = w_w \times (t_{w1} - t_{w2}) + 3.41\ E \qquad (38)$$

where w_w = water flow, lb per hr

t_{w1} = temperature of inlet water, F

t_{w2} = temperature of outlet water, F

E = electric power input to fan motor, watts

9.60 *Air Conditioner Using Electric Energy As Heating Medium*

9.61 The electric energy measurement method consists of measuring the electric power input to the resistance heaters. To this value should be added that portion of the electric power which represents heat added by the fan.

9.62 Heating capacity, q, in Btu per hr, is as follows:

$$q = 3.41\ E \qquad (39)$$

where E = total electric power input, watts

9.70 *Air Conditioners, Direct Fired*

9.71 Net total room heating effect on the heating medium side for direct fired air conditioners is determined by subtracting the stack, combustion, and jacket losses from the heat content of the fuel consumed.

9.72 It will be acceptable to use test apparatus and methods which are in common use, such as the American Gas Association test methods for gas fired equipment and the test methods of the Bureau of Standards for oil fired equipment.

9.73 If such methods are used, it may be impracticable to check the net heating effect within 6.0 percent as specified in Paragraph 9.01.

Section 10. HUMIDIFYING CAPACITY TESTS

10.00 *Requirement for Rating*

10.01 Humidifying capacity is determined by the quantity of water evaporated into the air per unit time. When air quantity is determined by the psychrometric calorimeter, it is also relatively easy to obtain a simultaneous test of humidifying capacity from the change in enthalpy of the air due to the moisture addition. The direct measurement of water evaporated shall be used as the rated capacity and the indirect method as a confirming test.

10.02 Humidifying capacity tests shall be made under conditions specified in Section 3, Tables 1 and 2.

10.03 Test instruments shall be as specified in Section 4.

10.04 Air flow measurements shall be as specified in Section 6.

10.05. Liquid flow measurements shall be as specified in Section 7.

10.06 Test conditions shall be maintained until equilibrium has been reached but for not less than one hour before recording data for rating test. The test shall then be run for at least one hour, recording data every ten minutes, giving at least seven sets of readings.

0.07 Table 8 lists required data. The table shows the general information required but is not intended to limit the data to be obtained.

Table 8. Data To Be Recorded for Humidifying Capacity Tests

m o.	*Item*	*Unit*
1	Date	—
2	Observers	—
3	Barometric pressure	in. Hg
4	Power consumption for each motor	watts
5	Applied voltage for each motor	volts
6	Temp of air surrounding air cond.	deg F
7	Dry bulb temp of room air entering unit	deg F
8	Wet bulb temp of room air entering unit	deg F
9	Dry bulb temp of air discharged by unit	deg F
0	Wet bulb temp of air discharged by unit	deg F
1	Flow rate of water to humidifier	lb/hr
2	Flow rate of water evap by humidifier	lb/hr
3	Flow rate of water to humidifier wasted	lb/hr

.10 *Calculations*

10.11 Direct measurement of humidifying capacity or water evaporation rate, w_w, in lb per hr, is determined from the following:

$$w_w = w_t - w_l \qquad (40)$$

where w_t = water flow rate to humidifier, lb per hr

w_l = water flow rate wasted from humidifier, lb per hr

10.12 The indirect method of determining humidifying capacity, w_w, in lb per hr, is calculated from the following:

$$w_w = 60 \frac{Q_n}{v_n} (W_2 - W_1) \qquad (41)$$

where Q_n = air flow through nozzle, cfm

v_n = specific volume of air at nozzle, cu ft per lb of dry air

W_2 = moisture content of air leaving unit, lb of water per lb of dry air

W_1 = moisture content of air entering unit, lb of water per lb of dry air

Section 11. DEFINITIONS

1.00 *Definitions*

11.01 *Accuracy of Readings* — Where percentage limits of readings are herein given, the reference basis is the magnitude of the greater absolute reading and not scale of instrument.

11.02 *Air Conditioner* is a specific air treating combination consisting of means for ventilation, air circulation, air cleaning, and heat transfer with control means for maintaining temperature and humidity within prescribed limits.

11.03 *Condensing Unit* is a specific refrigerating machine combination for a given refrigerant, consisting of one or more power driven compressors, condensers, liquid receivers (when required), and the regularly furnished accessories.

11.04 *Direct Fired* means the combustion of a fuel in an air conditioner with provision for directly heating air for circulation to the conditioned room.

11.05 *Evaporative Equilibrium of a Wet Bulb (wb) Instrument* is the temperature attained when the wetted wick has reached a stage of constant temperature. When the instrument is exposed to air at velocities over 1000 ft per min, this temperature may be considered a true wet bulb temperature.

11.06 *Forced Air Circulation* is that caused by a difference of pressure produced by a fan or blower.

11.07 *Free Delivery Type Unit* takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

11.08 *Humidifying Effect in Btu per Hour* is a latent heat at the average vaporizing temperature times the number of pounds of water evaporated per hour.

11.09 *Liquid Refrigerant Flowmeter* includes a mechanism and supplementary device for determining the instantaneous rate of circulation of liquid refrigerant.

11.10 *Liquid Refrigerant Quantity Meter* is an apparatus for determining the refrigerating capacity of a unit by measuring the circulation of refrigerant.

11.11 *Natural Convection Circulation* of air is that produced by gravity in response to a temperature difference of the air itself.

11.12 *Net Room Dehumidifying Effect* is the difference between the moisture content in lb per hr of the entering and leaving air multiplied by 1060 (Btu per lb), expressed in Btu per hr.

11.13 *Net Room Sensible Cooling Effect* is the difference between the net total cooling effect and the dehumidifying effect expressed in Btu per hr.

11.14 *Net Total Room Cooling Effect on an Air Conditioner* is the total useful capacity of the unit for removing heat from the space to be treated expressed in Btu per hr.

11.15 *Nonvolatile Refrigerant* is one which remains a liquid during the process of absorbing heat within the air conditioner.

11.16 *Pressure Type Unit* is one for use with one or more external elements which impose air resistance.

11.17 *Room Air Conditioner* is a factory made and cased assembly designed for location in the conditioned room, for free delivery of air (no discharge or return air ducts), and usually for installation in the window itself or in front of the window. It consists of a condensing unit and means for air circulation, ventilation, air cleaning, cooling, and dehumidification.

11.18 *Self-Contained Air Conditioner* is a factory made assembly of a condensing unit or absorption refrigeration unit and means for air circulation, ventilation, air cleaning, cooling, dehumidification, and control of temperature, and may include or have provision for the additional means of heating and humidifying.

11.19 *Sensible Heat Ratio* of an air conditioner is the ratio of the sensible cooling effect to the total cooling effect.

11.20 *Sensible Heating Effect* is the total useful capacity of the unit for adding sensible heat to the space being heated, expressed in Btu per hr.

11.21 *Standard Air* is air at a temperature of 68 F dry bulb and a standard barometric pressure of 29.92 in. of mercury, which contains 0.00725 lb of moisture per lb of dry air. Standard air weighs 0.07488 lb per cu ft.

11.22 *Standard Barometric Pressure* is a barometric pressure of 29.92 in. of mercury.

11.23 *Ventilation Air Capacity of a Unit* is the amount of air introduced to the room from an external source either directly or as a result of the exhaust of air from the room by the unit, expressed in cu ft per min of standard air.

11.24 *Volatile Refrigerant* is one which changes from the liquid to the vapor phase in the process of absorbing heat within the air conditioner.

Section 12. SYMBOLS

12.10 Primary symbols used in the computation of results of tests are as follows:

A = nozzle area, sq ft
B = volume, cu ft
c = specific heat of liquids, Btu per lb, F
C_d = nozzle discharge coefficient
D = diameter, in.
d = distance or height, in.
E = power, watts
f = volume of fluid per in. of cylindrical metering tank height, cu ft
H = total heat (enthalpy) of a mixture of air and water vapor, Btu per lb of dry air
h = velocity pressure of air at nozzle throat or static pressure difference across nozzle, in. of water
h_f = enthalpy of volatile refrigerant in liquid phase, Btu per lb
h_g = enthalpy of volatile refrigerant in gas phase, Btu per lb
h_{fg} = latent heat of vaporization, Btu per lb
K = a factor used for nozzle coefficient determination
M = weight, lb
P = barometric pressure, in. of mercury
ρ = density, lb per cu ft
Q = air volume flow rate, cfm
q = net heating effect, Btu per hr
q_d = net dehumidifying effect, Btu per hr
q_s = net sensible cooling effect, Btu per hr
q_h = net total heating effect, Btu per hr
q_t = net total cooling effect, Btu per hr
Σ = Sum of
R = Reynolds number
T = time, min
t = dry bulb temperature, F
t' = wet bulb temperature, F
V = velocity, fpm
v = specific volume of air and water vapor mixture at standard barometric pressure of 29.92 in. mercury, cu ft per lb of dry air
v' = specific volume of air and water vapor mixture, cu ft per lb of air-vapor mixture
W = moisture content of air, lb of water per lb of dry air
w = weight flow, lb per hr

12.20 Subscript symbols used in the computation of results of tests are as follows:

$_1$ = refers to entering state
$_2$ = refers to leaving state
$_a$ = refers to condenser air circuit of cooling devices or to heating air circuit of heat pumps
$_c$ = refers to cooling medium circuit of cooling devices or to heat absorption medium circuit of heat pumps
$_f$ = refers to refrigerant circuit, liquid phase
$_g$ = refers to refrigerant circuit, gas phase
$_i$ = refers to heat input
$_l$ = refers to loss
$_m$ = refers to calorimeter circuit
$_n$ = refers to nozzles
$_o$ = indicates standard air
$_r$ = refers to room air circuit
$_s$ = refers to steam
$_t$ = refers to total effects each as heating, cooling, humidifying, dehumidifying effects
$_v$ = refers to ventilation air circuit
$_w$ = refers to condenser water circuit of cooling devices, or to heating circuit of steam or water heaters or heat pumps, or to water evaporated in humidifier

ASRE Standard No. 17-R

ASRE Standard

METHOD OF RATING AND TESTING REFRIGERANT EXPANSION VALVES

Approved by The American Society of Refrigerating Engineers

December 8, 1948

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40th STREET NEW YORK 18, N. Y.

Price 50 Cents

Committee on Revision of ASRE Standard No. 17

F. Y. Carter, *Chairman*
B. E. James
A. B. Newton
P. B. Reed
A. L. Topp
D. D. Wile
J. E. Dube
W. F. Wischmeyer

ASRE Standards Committee, 1948

R. C. Jordan, *Chairman*

D. D. Wile, *Vice Chairman*
R. E. Backstrom
J. R. Chamberlain

H. M. Hendrickson
G. B. Priester
M. F. Tokach

ASRE Standard
Method of Rating and Testing Refrigerant Expansion Valves

SECTIONS

1. Scope and Purposes
2. Classification
3. Definitions
4. Method of Rating
5. Instruments
6. Apparatus
7. Method of Testing
8. Test Procedure
9. Data to Be Recorded
10. Examples

FOREWORD

e American Society of Refrigerating Engineers'
s provide standard methods for testing and rating
geration and air conditioning equipment, and/or set
tandard specifications for installation and perform-
of such equipment. This code may be used, in
whole or in part, by any association or government agency with due credit to The American Society of Refrigerating Engineers. Adherence is strictly on a voluntary basis and merely in the interest of obtaining uniform standards throughout the industry.

Section 1. SCOPE AND PURPOSES

Scope

1.11 This standard prescribes methods of rating and testing thermostatic and constant pressure expansion valves for use in refrigerant systems employing volatile refrigerants.

Purposes

1.21 The purposes of this Standard are:
- a. To list and define the terms used in rating
- b. To specify the data required
- c. To establish conditions and methods of test for obtaining standard ratings
- d. To specify the method of determining maximum pressure of thermostatic expansion valves provided with pressure limiting means

1.22 These standards are for rating only. They are not for production testing.

Section 2. CLASSIFICATION

Refrigerant expansion valves are classified as to type as follows:

2.11 *Constant Pressure*
- a. Internal equalizer
- b. External equalizer

2.12 *Thermostatic*
- a. Internal equalizer
- b. External equalizer

Section 3. DEFINITIONS

The following definitions apply only to parts and terms used in this standard:

3.01 *Thermostatic Expansion Valve*—A controlling device for regulating the flow of volatile refrigerant into a cooling unit, actuated by changes in superheat of the refrigerant vapor leaving the cooling unit.

3.02 *Constant Pressure Expansion Valve*—A controlling device for regulating the flow of volatile refrigerant into a cooling unit, actuated by changes in pressure of the low side refrigerant.

3.03 *Capacity of an Expansion Valve*—The refrigerating effect in Btu per hour, or tons, each of 12,000 Btu per hour, produced by the evaporation of refrigerant passed by the valve under specified conditions.

3.04 *Superheat*—The difference between the temperature of the thermal bulb and the temperature corresponding to the pressure at the outlet, or at the equalizer connection when provided, of a thermostatic expansion valve.

3.05 *Superheat Change*—The change in superheat of a thermostatic expansion valve required to open the valve a predetermined amount.

3.06 *Pressure Change*—The change in outlet pressure of a constant pressure expansion valve required to open the valve a predetermined amount.

3.07 *Liquid Refrigerant Quantity Meter*—A device for determining by volume or weight a quantity of refrigerant.

3.08 *Volatile Refrigerant*—A substance which changes from the liquid to the vapor phase in the process of absorbing heat within the evaporator.

3.09 *Liquid Refrigerant Flowmeter*—A device for determining the rate of flow of a refrigerant.

3.11 *Internal Equalization*—Where the expansion valve pressure element responds to the pressure leaving the valve outlet.

3.12 *External Equalization*—Where the expansion valve pressure element responds to the pressure from any remote location, said pressure communicated to valve by means of a pressure line external to the valve.

3.13 *Certified Standard*—An instrument calibrated by the manufacturer or other reliable agency that is designed as a standard to which all similar instruments can be compared and is certified as to accuracy.

Section 4. METHOD OF RATING

4.10 *Rating*

4.11 The capacity of an expansion valve shall be expressed in Btu per hour or in tons. It is recommended that published ratings be given in tons.

4.12 The rating of thermostatic expansion valves shall include all the conditions listed in Table 1 for which the valve is recommended, it being understood that ratings may also be given for superheat changes exceeding 7 deg F if specified as such.

Table 1.

Condition	*Saturation temp, F, at valve outlet*	*Saturation temp, F, at valve inlet*	*Superheat change, deg F*
A	—40	100 F Vapor free liquid	7 or less
B	—10		
C	+20		
D	+40		

4.13 The rating of constant pressure expansion valves shall include all the conditions listed in Table 2 for which the valve is recommended, it being understood that ratings may also be given for pressure changes exceeding 4 lb if specified as such.

Table 2.

Condition	*Saturation temp, F, at valve outlet*	*Saturation temp, F, at valve inlet*	*Pressure change, psi*
A	—40	100 vapor free liquid	4 lb or less
B	—10		
C	+20		
D	+40		

4.14 Nominal rating shall be given under at least one of the conditions specified in Table 1 or 2.

4.15 The liquid entering the expansion valve must be vapor free, but shall not be subcooled more than 10 deg F. A correction must be made for this subcooling when arriving at the actual rating. The correction for subcooling automatically results: if the *flow rate* is calculated using the actual liquid temperature; and the *effective cooling per pound of refrigerant circulated* is calculated using the standard specified liquid temperature as per the examples in Section 10.

4.20 *Data Required in Standard Ratings*

4.21 The rating of an expansion valve shall include all the following data which apply to its function.

- a. Name of refrigerant
- b. Temperature and pressure of volatile refrigerant entering the expansion valve
- c. Temperature and pressure of volatile refrigerant leaving the expansion valve
- d. Temperature of sensitive element
- e. Quantity of refrigerant circulated per hour
- f. No-flow superheat—thermostatic valves
- g. Superheat change—thermostatic
- h. Pressure change—constant pressure valves
- i. Pressure change from zero to rated cap city on thermostatic valves with pressu limiting features, temperature sensiti element being held at 80 F.

4.30 *Rating Tolerance*

4.31 Due to manufacturing tolerance, the norm capacity of a valve shall not be less than t rated capacity.

4.40 *Published Ratings*

4.41 Published ratings shall include ratings one or more of the conditions specified Tables 1 and 2 and shall be properly ide tified.

Section 5. INSTRUMENTS

5.10 Instruments shall be of the types and accuraci listed below and shall be calibrated against cer fied standards.

5.20 *Temperature Measuring Instruments*

5.21 Temperature measurements shall be ma with one or more of the following instr ments.

- a. Glass thermometers
- b. Thermocouples
- c. Electric resistance thermometers.

5.22 Accuracy shall be within the following limit

- a. Expansion valve temperature-sensiti element temperature ± 0.2 deg F.
- b. All other temperatures ± 0.5 deg F.

5.23 In no case shall the smallest scale divisi of the measuring instrument exceed tw times the specified accuracy.

5.30 *Pressure Measuring Instruments*

5.31 Pressure measurements shall be made wi one or more of the following instruments:

- a. Mercury column
- b. Bourdon tube gage

5.32 Accuracy shall be within the followin limits:

- a. Outlet pressure, absolute, ±1.0 %
- b. Inlet pressure, absolute, ± 2.0 %

5.33 Calibration of a bourdon tube gage shall with respect to a dead weight tester or comparison to a mercury column.

5.34 In no case shall the smallest scale division the measuring instrument exceed 2½ tim the specified accuracy.

5.40 *Refrigerant Flow Measuring Instruments*

5.41 Flow measurements shall be made with o or more of the following instruments, ha ing an accuracy of within ± 3% of t quantity measured.

- a. Liquid refrigerant quantity meter mea ing either weight or volume
- b. Liquid refrigerant flow meter
- c. Calibrated condensing unit as defined ASRE Circular No. 14
- d. Condenser water method as defined ASRE Circular No. 14
- e. Calibrated suction line orifice or nozzle

Calorimeters must be of the type and accuracy defined in ASRE Circular No. 14 and all temperature, pressure, electrical, brine or water flow measurements must be within the specified limits.

Other Measurements

5.61 Time measurements shall be made with an accuracy of within ± 0.2%

5.62 Weight measurements shall be made with an accuracy of within ±0.2%

Section 6. APPARATUS

The expansion valve to be tested or rated shall be installed in a refrigerating system composed of essential elements as follows:

6.11 One or more condensing units so arranged that variable capacities can be obtained

6.12 Suitable calorimeters or liquid flowmeters for determining the quantity of refrigerant passing through the expansion valve

6.13 Pressure and temperature measuring instruments

6.14 Constant temperature bath for maintaining the temperature of the temperature sensitive element of thermostatic expansion valves

6.15 Sight glasses for determining the quality of liquid entering the expansion valve and for determining when flow through the valve ceases. The opening in the sight glasses shall not be less than the size of the valve connections.

6.16 Fittings for inserting sensitive element of temperature measuring instruments into refrigerant tubes

6.17 Hand-operated shutoff valves

6.18 Hand-operated expansion valves

The expansion valve shall be installed in the refrigerating system in the manner shown in Figure 1.

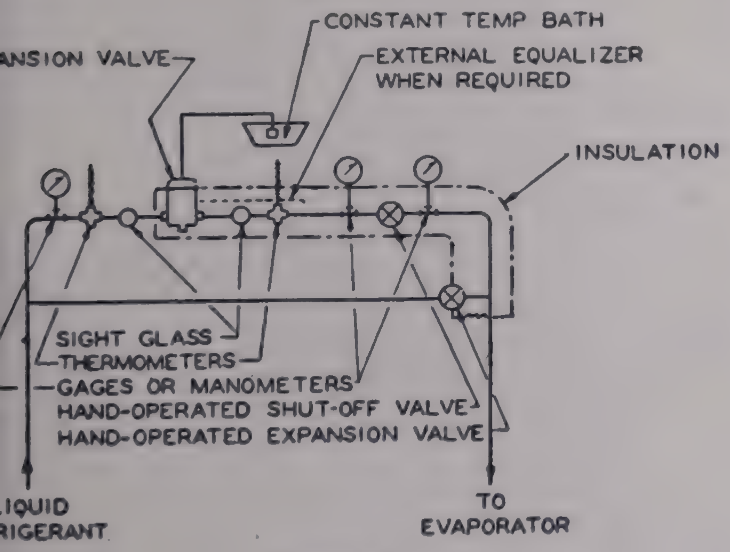


FIGURE 1.

Section 7. METHOD OF TESTING

Basis of Rating

7.11 The expansion valve shall be open an amount equivalent to the superheat change or pressure change recommended by the manufacturer.

7.12 Rated opening shall be approached with the valve moving from the closed to the open position.

7.20 Refrigerant flow test shall consist of determination of cooling effect by one of the methods listed in Paragraph 5.41.

Section 8. TEST PROCEDURE

8.10 *Thermostatic Expansion Valves*

The flow rate through a thermostatic expansion valve is a function of the needle opening (distance needle has moved away from seat). The position of the needle with reference to the seat is determined by the superheat of the valve and the no-flow superheat adjustment. The superheat of a thermostatic expansion valve is the difference between the temperature of the thermal bulb and the temperature corresponding to the pressure at the valve outlet, or equalizer connection when used. Superheat may be changed by varying either the temperature of the thermal bulb or the pressure at the valve outlet while holding the other constant. For convenience in testing, and in the following description, the thermal bulb temperature is held constant and the superheat is varied by changing the pressure at the valve outlet.

If the valve is closed and the superheat is increased (outlet pressure decreased with thermal bulb held at constant temperature), a value of superheat will be reached at which the valve will just begin to open. If the valve is open and the superheat is decreased (outlet pressure increased with thermal bulb held at constant temperature), a value of superheat will be reached at which the valve will just close. These two superheats may be different due to friction in the expansion valve mechanism. In order to penalize an expansion valve that has excessive friction, the *no-flow superheat* is taken as the superheat at which the valve closes when approaching from a high superheat or valve open position and is indicated in Figure 2 as the difference between a bulb temperature of 55 and a suction temperature of 45 F. As the superheat is increased from the no-flow superheat setting, the needle approaches the rated flow position. The superheat at rated flow is the *operating superheat* and is indicated in Figure 2 as the difference between a bulb temperature of 55 F and a suction temperature of 40 F.

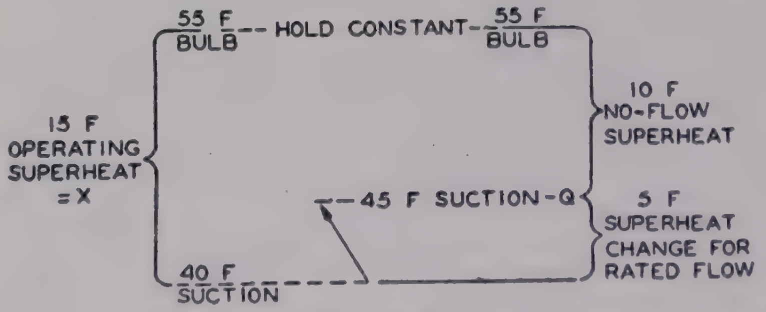


FIGURE 2.

The difference between the *operating superheat* and the *no-flow superheat* is the *superheat change* and is the basis on which the thermostatic expansion valve is rated.

This test procedure, then, is for the purpose of establishing the actual test condition which will result in the recommended superheat change.

8.11 The test setup shall be essentially as shown in Figure 1.

8.12 Let X be the operating superheat of the valve at rated capacity, determined by adding the superheat setting to the specified superheat change.

a. Maintain the expansion valve temperature sensitive element at X degrees F above the valve outlet refrigerant temperature at which the valve is to be rated.

b. The inlet liquid pressure shall correspond to a 100 F saturation temperature.

c. Solid liquid shall be maintained to inlet of expansion valve as indicated by sight glass, but subcooling shall not exceed 10 F

8.13 Adjusting the test equipment:

a. Close hand-operated shutoff valve

b. Adjust the hand-operated expansion valve to obtain an evaporator pressure slightly lower than the pressure corresponding to the outlet temperature at which the valve is to be rated.

8.14 Check the superheat setting of the valve as follows:

a. Slowly open the hand-operated shutoff valve until refrigerant flow through the thermostatic expansion valve commences as observed in the outlet sight glass.

b. After flow is established, slowly close the hand-operated shutoff valve and determine the outlet pressure at which flow ceases as observed in the outlet sight glass.

c. Let Q degrees F be the saturation temperature of the outlet pressure at which flow ceases.

d. The no-flow superheat setting is the bulb temperature minus Q degrees F.

e. The no-flow superheat setting shall not vary from the specified setting by more than ±2 deg F. If the superheat setting is not specified, 10 deg F superheat setting shall be used.

8.15 Let X' equal the sum of the no-flow superheat just measured and the specified superheat change.

a. The expansion valve temperature sensitive element shall hereafter be maintained at X' degrees F above the valve outlet refrigerant temperature at which the valve is to be rated.

8.16 *Capacity Test*

a. With hand shutoff valve fully open and hand expansion valve closed adjust the flow rate to establish the expansion valve outlet pressure corresponding to the temperature at which the valve is to be rated. The pressure change from the no-flow to the rated flow position shall not exceed the pressure equivalent of the specified superheat change.

b. Whenever it is necessary to adjust the testing equipment, always make adjustments in a manner which will allow the expansion valve needle to move from the closed to the open position. This shall be accomplished by closing hand shutoff valve, causing pressure to increase and move expansion valve needle to closed position. Then repeat procedure as d
scribed in Paragraph 8.16a.

c. Run capacity test following in gene
the practice of ASRE Circular No. 14.

d. The test shall be continued until four su
cessive readings of capacity data taken
not less than 10 minute intervals do n
vary by more than 5%, the minimum
the four readings being the basis of co
parison.

e. The average value of the four readin
is to be used in the capacity calculation.

8.17 Check the pressure limiting means of the
mostatic expansion valves having this fe
ture as follows:

a. Hold expansion valve temperature sen
tive element at 80 F, ±1 deg F.

b. Close hand-operated shutoff valve.

c. Adjust the hand-operated expansion val
to obtain an evaporator pressure sligh
lower than the maximum pressure in
cated on the valve.

d. Slowly open the hand-operated shut
valve until refrigerant flow through t
thermostatic expansion valve commenc
as observed in the outlet sight glass a
carefully note the valve outlet pressu

e. Close hand-operated expansion valve a
adjust the flow rate to establish a val
outlet pressure approximately 2 psi und
the pressure at which the valve initia
opened.

f. Run a capacity test following in gene
Paragraph 8.16.

g. Repeat above procedure several tim
each subsequent test being run at appro
mately 2 psi lower valve outlet pressure

h. Plot a curve of Capacity vs. Outlet Pr
sure from the data obtained in Paragra
8.17f and 8.17g. This curve, when used
conjunction with a condensing unit ca
city curve, will indicate the maxim
pressure at which the valve will bala
the particular condensing unit dema

8.18 A sample of the liquid refrigerant shall
taken during the capacity test to determi
the amount of oil present.

a. The quantity of oil circulating shall n
exceed 2% by weight.

b. Use oil separators, if necessary, or ot
means to reduce the oil being circulat

8.20 *Constant Pressure Expansion Valves (Au matic Expansion Valves)*

The flow rate through a constant pressure exp
sion valve is a function of the needle open
(distance needle has moved away from seat). T
position of the needle with reference to the seat
determined by the pressure at the valve out
and the no-flow pressure (setting) of the val
If the valve is closed and the outlet pressure
decreased, a pressure is reached at which the va
just begins to open. If the valve is open and
outlet pressure is increased, a pressure is reach
at which the valve just closes. These t
pressures may be different due to friction in
expansion valve mechanism. In order to penal
a valve that has excessive friction, the *no-fl*
pressure setting is the pressure at which the va
closes when approaching from a lower pressure
valve open position and is indicated in Figu

as "no-flow pressure setting = Q." As the outlet pressure is decreased, the needle approaches the rated flow position. The pressure at rated flow position is indicated in Figure 3 as *operating pressure setting.*

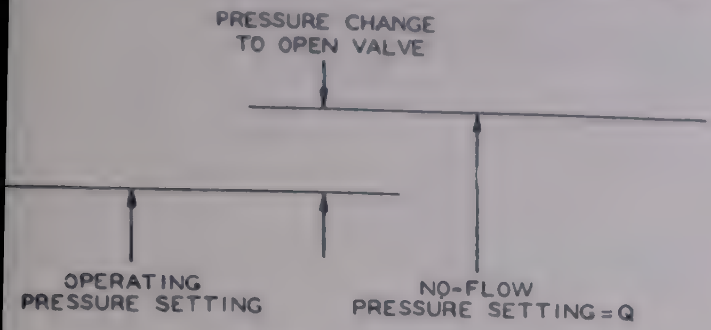


FIGURE 3.

The difference in pressure between the *operating* and *no-flow pressure setting* is the *pressure change* and is the basis on which the constant pressure expansion valve is rated.

This test precedure, then, is for the purpose of establishing the actual test conditions which will result in the recommended pressure change.

8.21 The testing arrangements shall be essentially as shown in Figure 1. The constant temperature bath is not required in testing constant pressure expansion valves.

8.22 Let Q be the pressure determined by adding the specified pressure change to the pressure corresponding to the outlet temperature at which the valve is to be rated.

8.23 Adjusting the test equipment:
- a. Close the hand-operated shutoff valve.
- b. Adjust the hand-operated expansion valve to obtain an evaporator pressure slightly lower than Q psi.

8.24 Check the valve setting as follows:
- a. Slowly open the hand-operated shutoff valve and adjust the constant pressure expansion valve until refrigerant flow through the valve commences as observed in the outlet sight glass.
- b. After flow is established, slowly close the hand-operated shutoff valve and determine the valve outlet pressure at which flow ceases as observed in the outlet sight glass. This pressure should be Q psi.
- c. If this pressure differs from Q by more than ± 1 lb, the valve should be readjusted and rechecked as outlined above.
- d. The setting having been established, the valve is not to be further adjusted throughout the capacity test.

8.25 *Capacity Test*
- a. With hand shutoff valve fully open and hand expansion valve closed, adjust the testing equipment to increase the refrigerant flow rate gradually. This will cause the valve outlet suction pressure to decrease. When the pressure has decreased by the specified pressure change, the valve is opened the rated amount. The actual pressure change shall not exceed that specified by the manufacturer for rated capacity.
- b. Whenever it becomes necessary to adjust the testing equipment, always make adjustment in a manner which will allow the expansion valve needle to move from the closed to the open position. This shall be accomplished by closing hand shutoff valve, causing pressure to increase and move expansion valve needle to closed position. Then repeat procedure as described in Paragraph 8.25a.
- c. Run capacity test following in general the practices of ASRE Circular No. 14.
- d. The test shall be continued until four successive readings of capacity data taken at not less than 10 minute intervals do not vary by more than 5%, the minimum of the four readings being the basis of comparison.
- e. The average value of the four readings is to be used in the capacity calculation.

8.26 A sample of the liquid refrigerant shall be taken during the capacity test to determine amount of oil present.
- a. The quality of oil circulating shall not exceed 2% by weight.
- b. If necessary, use oil separators or other means to reduce the oil being circulated.

Section 9. DATA TO BE RECORDED

9.10 *General Data*

9.11 Date
9.12 Observers
9.13 Barometric pressure

9.20 *Data of Expansion Valve*

9.21 Manufacturer's name and address
9.22 Size, type, and designation
9.23 Name of refrigerant
9.24 Specified superheat change—thermostatic expansion valve
9.25 Specified pressure change—constant pressure expansion valve

9.30 *Refrigerant Flow Data*

9.31 *Thermostatic Expansion Valve*
- a. Temperature and pressure of liquid entering expansion valve
- b. Temperature and pressure of refrigerant leaving expansion valve
- c. Expansion valve bulb temperature
- d. No-flow superheat
- e. Operating superheat during test
- f. Other data required for determining capacity shall be recorded as stated in ASRE Circular No. 14 for the particular test method used.

9.32 *Automatic Expansion Valve*
- a. Temperature and pressure of liquid entering expansion valve
- b. Temperature and pressure of refrigerant leaving expansion valve
- c. No-flow pressure
- d. Operating pressure
- e. Other data required for determining capacity shall be recorded as stated in ASRE Circular No. 14 for the particular method used.

Section 10. EXAMPLES

10.10 *Thermostatic Expansion Valves*

10.11 The equipment is arranged as shown in Fig. 1.

10.12 The barometric pressure shall be taken as 29.92 in. Hg and all pressure readings corrected accordingly.

10.13 Assume the valve manufacturer's specified superheat setting to be 10 deg F and the specified superheat change to be 5 deg F. Therefore, $X = 15$. Assuming that the valve is to be rated at 40 F valve outlet refrigerant temperature, the expansion valve temperature sensitive element should be maintained at 40 plus 15 = 55 F.

10.14 Assuming that a Freon-12 valve is being tested, the valve outlet pressure for the test is 37.0 psi corresponding to 40 F saturation. Close the hand-operated shutoff valve. Adjust the hand-operated expansion valve to obtain a pressure several pounds below 37.0 psi in the evaporator.

10.15 Slowly open the hand-operated shutoff valve until refrigerant flow through the thermostatic expansion valve commences as observed in the valve outlet sight glass. Then slowly close the hand-operated shutoff valve and determine the valve outlet pressure at which flow ceases as observed in this sight glass. Assume that this pressure reading is found to be 42.7 psi. The temperature Q corresponding to this saturated refrigerant pressure is 46 F. The actual no-flow superheat setting of the test valve is the difference between the temperature sensitive element and Q.
Actual no-flow superheat setting = 55 — 46 = 9 F.

10.16 Let X^1 degrees F = actual no-flow superheat setting plus the specified superheat change = 9 + 5 = 14 F.
The expansion valve temperature sensitive element should hereafter be maintained X^1 degrees F above the temperature at which the valve is to be rated.
Thus, temperature sensitive element temperature = 40 + 14 = 54 F

10.17 Establish balanced conditions with valve outlet pressure corresponding to saturated refrigerant temperature of 40 F with the hand-operated shutoff valve open and with the hand-operated expansion valve closed. Run a capacity test. Continue the test until four successive readings are obtained as specified in Paragraph 8.16d.

10.18 While the equipment is still functioning under the balanced conditions of the capacity test, check the percentage of oil by weight in the liquid refrigerant entering the expansion valve as covered in Paragraph 8.18.

10.19 *Test results and calculations*

a. Calorimeter Method
Test Results:

Valve outlet refrigerant	40 F 37.0 psig
Valve inlet refrigerant	98 F 117.0 psig
Calorimeter measurement	3000 Btu per hr

Calculations are as follows:
Enthalpy for:

Saturated liquid at 100 F $= h_{100}$
$h_{100} = 31.1$ Btu per lb

Saturated liquid at 98 F $= h_{98}$
$h_{98} = 30.7$ Btu per lb

Saturated vapor at 40 F $= H_{40}$
$H_{40} = 82.7$ Btu per lb

Refrigerating effect in test
$= H_{40} — h_{98}$
$= 82.7 — 30.7$
$= 52.0$ Btu per lb

Refrigerant circulated
$= 3000/52$
$= 57.7$ lb per hr

Refrigerating effect at standard conditions
$= H_{40} — h_{100}$
$= 82.7 — 31.1$
$= 51.6$ Btu per lb

Capacity of expansion valve at standard conditions
$= 57.7 \times 51.6$
$= 2977$ Btu per hr

b. Flowmeter Method
Test Results:

Valve outlet refrigerant	40 F 37.0 psig
Valve inlet refrigerant	98 F 117.0 psig
Flowmeter measurement	57.7 lb per hr

Calculations are as follows:
Enthalpy for:

Saturated liquid at 100 F $= h_{100}$
$h_{100} = 31.1$ Btu per lb

Saturated vapor at 40 F $= H_{40}$
$H_{40} = 82.7$

Refrigerating effect at standard conditions
$= H_{40} — h_{100}$
$= 82.7 — 31.1$
$= 51.6$ Btu per lb

Capacity of expansion valve at standard conditions
$= 57.7 \times 51.6$
$= 2977$ Btu per hr

10.20 *Thermostatic Expansion Valve — Limitin Pressure Means*

10.21 The equipment is arranged as shown i Figure 1.

10.22 The barometric pressure shall be taken a 29.92 in. Hg and all pressure reading corrected accordingly.

19.23 With expansion valve bulb held at 80 and hand-operated shutoff valve closed, a just hand-operated expansion valve to o tain a pressure in the evaporator sever pounds below the maximum pressure ind cated on the valve.

10.24 Slowly open the hand-operated shuto valve until refrigerant flow through th thermostatic expansion valve commenc as observed in the valve outlet sight glas Then slowly close the hand-operated shu off valve and determine the valve outl pressure at which flow ceases as observe in this sight glass. Assume that th pressure reading is found to be 58 psi.

10.25 Establish balanced conditions with valv outlet pressure 2 psi lower or 56 psi, wit hand-operated shutoff valve open and han operated expansion valve closed. Run capacity test. Continue the test until fou successive readings are obtained as spec fied in Paragraph 8.16d.

10.26 Establish balanced conditions with outl pressure 2 additional psi lower or 54 psi, and repeat capacity test as per Paragraph 10.25.

10.27 Repeat this procedure, conducting each successive test at 2 psi lower pressure than the immediately preceding test, until a capacity is obtained which exceeds the intended capacity at which the valve will be used.

10.28 *Test Results and Calculations*

a. Follow the procedure shown in Paragraph 10.19a and calculate valve capacity for each test.

b. Assume the test data results as follows: (This procedure automatically penalizes a valve having excessive friction.)

Outlet pressure, psi	*Valve cap., Btu per hr*
58	0
56	2000
54	6000
52	10000

c. Plot curve of Valve Capacity vs. Valve Outlet Pressure as indicated in Figure 4.

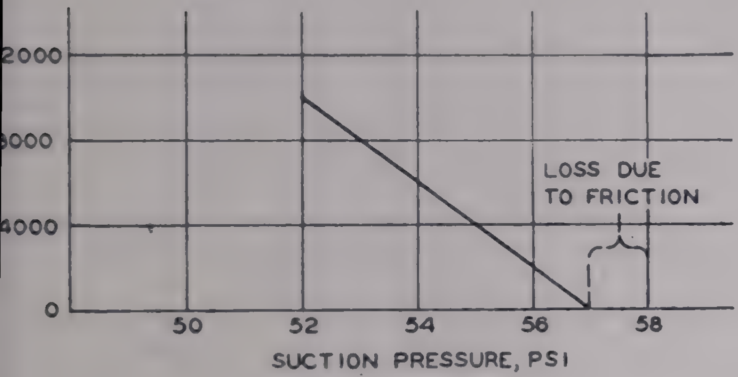


FIGURE 4.

.30 *Constant Pressure Expansion Valves*

10.31 The equipment is arranged as shown in Figure 1.

10.32 Assume conditions to be as follows:

a. Refrigerant: Freon-12

b. Capacity rating to be at 40 F
The pressure corresponding to 40 F is 37.0 psig
Specified pressure change = 4 psi
Q = valve closing pressure = 37.0 + 4 = 41.0 psig
Throughout the following test procedure the valve inlet pressure corresponding to saturated refrigerant temperature of 100 F is to be maintained.

10.33 The constant pressure expansion valve is now to be adjusted to close off at approximately 41 psig. This is accomplished in the following manner: First, by means of the adjusting stem the constant pressure expansion valve is set to close off at some pressure lower than 41 psig. Second, with the hand-operated shutoff valve open, the hand-operated expansion valve is adjusted to give a pressure of 41 psig. Third, the constant pressure expansion valve is adjusted by means of the adjusting stem until flow through the valve commences as observed in the outlet sight glass.

10.34 The exact closing pressure of the constant pressure expansion valve is now determined. The hand-operated shutoff valve is closed and the hand operated expansion valve is adjusted to give a low side pressure of approximately 38 psig. The hand-operated shutoff valve is slowly opened until flow through the valve commences as observed in the sight glass. Then the hand-operated shutoff valve is slowly closed and the pressure at which flow ceases is observed by means of the sight glass and found to be 41 psig. No further adjustment is made on the constant pressure expansion valve.

10.35 With the hand-operated shutoff valve open, and the hand-operated expansion valve closed, the test equipment is now adjusted to obtain a low side pressure of 37 psig, corresponding to 40 F saturation temperature. A capacity test is run and continued until four successive readings within the specified limits are obtained, as specified in Paragraph 8.25d.

NOTE. It is important that the valve opening pressure—i.e., the difference between the close-off pressure and the low side test pressure—does not exceed four lb.

10.36 While the equipment is still functioning under the balanced conditions of the capacity test, check the percentage of oil by weight in the liquid refrigerant entering the expansion valve as covered in Paragraph 8.26.

10.37 *Test Results and Calculations*

a. Calorimeter Method

Test results:

Valve outlet refrigerant	40 F 37.0 psi
Valve inlet refrigerant	98 F 117.0 psi
Calorimeter measurement	3000 Btu per hr

Calculations are as follows:
Total heat above —40 F
Enthalpy for:
Saturated liquid at 100 F = h_{100}
h_{100} = 31.1 Btu per lb
Saturated liquid at 98 F = h_{98}
h_{98} = 30.7 Btu per lb
Saturated vapor at 40 F = H_{40}
H_{40} = 82.7 Btu per lb
Refrigerating effect in test = $H_{40} - h_{98}$
= 82.7 — 30.7
= 52.0 Btu per lb
Refrigerant circulated = 3000/52
= 57.7 lb per hr
Refrigerating effect at standard conditions = $H_{40} - h_{100}$
= 82.7 — 31.1
= 51.6 Btu per lb
Capacity of expansion valve at standard conditions = 57.7 × 51.6
= 2977 Btu per hr

b. Flowmeter Method

Test Results:

Valve outlet refrigerant	40 F 37.0 psi
Valve inlet refrigerant	98 F 117.0 psi
Flowmeter measurement	0.961 lb per min

Calculations are as follows:

Total heat above —40 F

Enthalpy for:

Saturated liquid at 100 F = h_{100}
h_{100} = 31.1 Btu per lb
Saturated vapor at 40 F = H_{40}
H_{40} = 82.7

Refrigerant circulated = 60 × 0.961
= 57.7 lb per hr

Refrigerating effect at standard conditions = $H_{40} - h_{100}$
= $82.7 - 31.1$
= 51.6 Btu per lb

Capacity of expansion valve at standard conditions = 57.7 × 51.6
= 2977 Btu per hr

ASRE STANDARD No. 18

ASRE Standard

METHODS OF RATING AND TESTING F-CONTAINED MECHANICALLY REFRIGERATED DRINKING WATER COOLERS

Approved by The American Society of Refrigerating Engineers

January 19, 1940

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40TH STREET NEW YORK 18, N. Y.

Price 35 Cents

Special A S R E Standards Subcommittee on Water Coolers

J. L. Gibson, *Chairman*
A. A. Ralston
R. H. Tull
J. J. Corey
H. F. Hutzel

A S R E Standards Committee

Chester Lichtenberg, *Chairman*

Glenn Muffly	L. L. Lewis
H. D. Edwards	H. E. Keeler
L. S. Morse	W. R. Woolrich

Standard Methods of Rating and Testing Self-Contained Mechanically Refrigerated Drinking Water Coolers

DEFINITIONS

0 *A Drinking Water Cooler* consists of an assembly ich either employs or is used in conjunction with a me- anical condensing unit for the purpose of cooling drink- g water.

0 *A Self-Contained Drinking Water Cooler* consists es- tially of a cooling unit or primary and secondary cool- g units for cooling drinking water, a mechanical condens- g unit which may be air-cooled or water-cooled or air d water cooled, a device for drawing the cooled water d a drain for the unconsumed water, all enclosed in the me case or container.

1 *Types* of self-contained drinking water coolers are,
Bottled Water Cooler
Faucet Pressure Water Cooler
Bubbler Pressure Water Cooler.

2 *Bottled Water Cooler* is one which employs a bottle for ring the supply of water to be cooled and which utilizes faucet for filling glasses or cups.

3 *Faucet Pressure Water Cooler* is one which is uipped to use water under pressure from a piped system a supply to the cooler and which utilizes a faucet for ling glasses or cups.

24 *Bubbler Pressure Water Cooler* is one which is uipped to use water under pressure from a piped system a supply to the cooler and which employs a valve in the inking water line for controlling the flow of water to a bbler or open flowing stream so that the water may be unk without utilizing glasses or cups.

25 *Waste Water* is used in some bubbler pressure water olers in one of the following ways:
a. To precool the water flowing into the cooling unit by transferring heat from this water to the waste water by means of a heat interchanger or precooler.
b. To assist in cooling the refrigerant condenser or condensed refrigerant in addition to or instead of precooling the incoming drinking water.

30 *Capacity* of a drinking water cooler is the quantity water it will cool in a given ambient temperature with a ven ingoing water temperature and a given outgoing ater temperature.

40 *A. S. R. E. Standard Capacity Rating* of a drinking ater cooler is the number of gallons per hour cooled der the conditions specified in Table I.

50 *A. S. R. E. Standard Rating Conditions* for drinking ater coolers are given in Table I.

Table I. Rating Conditions

	Temperature in Deg. F.					*Amount of precooler water used in per cent of total drinking water drawn*
	Amb-ient	*Cooling water*		*Drinking water*		
ype of Cooler		*In-going*	*Out-going*	*In-going*	*Out-going*	
ottled water	90	—	—	90	50	
aucet pressure	90	80	100	80	50	
ubbler pressure						
No precooling	90	80	100	80	50	
With precooling	90	80	100	80	50	60

1.60 *A. S. R. E. Standard Power Input Rating* of a drinking water cooler is its total power input in watts when the cooler is operated under the conditions specified in Table I.

1.70 *A. S. R. E. Standard Performance Factor* of a drinking water cooler is the ratio of its energy input to capacity expressed in kilowatt-hours per gallon of water cooled.

1.80 *A. S. R. E. Standard Water Consumption Rating* of a drinking water cooler is the total number of gallons per hour required for cooling the condensing unit when it is operated under the conditions specified in Table I.

TESTING

2.10 *Test* of A. S. R. E. Standard Capacity shall be conducted by drawing, at a uniform rate, the maximum amount of drinking water possible under the rating conditions specified in Table I, while measurements are made of drinking water flow, power input, and cooling water consumption.

2.20 *Instruments*, whose types and accuracies are listed below, shall be used during and calibrated against secondary standards before and after each test.

2.21 *Temperature measuring instruments.*
  2.22 Temperature measurements shall be made with one or more of the following instruments:
    a. Mercury-in-glass thermometers.
    b. Thermocouples.
    c. Electric resistance thermometers.
  2.23 Accuracy shall be within the following limits:
    a. Drinking water, plus or minus 0.1°F.
    b. All other temperatures, plus or minus 0.5°F.

2.31 *Pressure measuring instruments.*
  2.32 Pressure measurements when taken shall be made with one or more of the following instruments:
    a. Mercury column.
    b. Bourdon tube gage.
  2.33 Accuracy shall be within the following limits:
    a. Suction pressure (absolute) plus or minus 1 lb. per sq. in.
    b. Other pressures (absolute) plus or minus 2 lb. per sq. in.

2.41 *Electrical measuring instruments.*
  2.42 Electrical measurements shall be made with one or more of the following instruments:
    a. Indicating.
    b. Integrating.
  2.43 Accuracy shall be within the following limits:
    a. Indicating instruments within 0.5 per cent of full scale rating.
    b. Integrating instruments within 1 per cent.

2.51 *Water flow measuring instruments.*
  2.52 Volume measurements shall be made with one or more of the following instruments:
    a. Liquid quantity meter, measuring either volume or weight where standard water weighs 8.38 pounds per gallon.
    b. Liquid flow meter.
  2.53 Accuracy shall be within plus or minus 1 per cent.

2.61 *Time measurements* shall be made with apparatus whose accuracy is within plus or minus 0.1 per cent.

2.71 *Weight measurements* shall be made with apparatus whose accuracy is within plus or minus 0.2 per cent.

3.10 Preparation

3.11 The drinking water cooler with all panels in place shall be tested in a room in which the temperature can be controlled. Panels shall remain in place throughout the entire test.

3.12 Pressure water coolers shall have a thermometer well attached to the inlet of the drinking water cooler close to the point of entrance of the supply line.

3.13 Pressure water coolers shall have an adjustable automatic pressure regulating valve located in the drinking water inlet line. This shall be connected to a pressure water supply which is provided with a means of controlling the water temperature.

3.14 Pressure bubbler water coolers shall have a thermometer well and hand regulated shutoff valve attached at the cooled water outlet in place of the bubbler head. The hand regulated shutoff valve shall be of a type suitable for closely controlling the rate of flow. Any piping between the thermometer well and water cooler enclosure shall be well insulated.

3.15 Faucet water coolers shall be tested with the faucet in place.

3.16 Bottle water coolers shall be tested with the largest size bottle for which the cooler is designed.

3.17 Water cooled drinking water coolers shall have thermometer wells located outside the water coolers but as near as practicable to the condensing unit cooling water inlet and outlet connections and the cooling water inlet shall be connected to a pressure water supply which is provided with a means of controlling the water temperature.

3.18 Drinking water coolers shall have their condensing units operated for a total of not less than 72 hr. before the test is started.

3.19 Locate the water cooler being tested in a room so that its temperature will not be affected by direct radiation to or from external cooling or heating equipment. The air circulation in the room shall be such that the specified uniformity of ambient temperature is obtained without direct draft upon the water cooler being tested. The windows of the test room shall be provided with suitable radiation shields and the water cooler shall be shielded from forced air currents which have a velocity in excess of 50 ft. per min.

4.10 Procedure

4.11 Start the condensing unit and regulate the voltage at the motor service connection to within plus or minus 2 per cent of the motor rated voltage.

4.12 Air cooled drinking water coolers shall have no abnormal air flow created over the condensing unit.

4.13 Water cooled drinking water coolers shall have the temperature of the inlet water flowing to the refrigerant condenser of the condensing unit adjusted to 80°F. plus or minus 0.5°F., and the water rate adjusted to give an outlet temperature of 100°F. plus or minus 0.5°F. No abnormal air flow shall be created over the condensing unit.

4.14 Pressure bubbler water coolers equipped with a precooler and when being tested for capacity with the precooler shall have the outlet hand regulated shutoff valve arranged to by-pass 60 per cent of the total stream to the drain and 40 per cent to an outside drain. The flow from both streams shall be noted and their sum reported as "Gallons of Drinking Water Per Hour."

4.15 Adjust the inlet drinking water temperature from pressure water systems or the average temperature of the water in the bottles for bottle water coolers to within plus 0.3°F. and minus 0.0°F. of the rating values specified in Table I.

4.16 For bubbler-type water coolers rated four (4) gallons per hour or greater, adjust the temperature of the outlet drinking water to 50°F. between plus 0.0°F. and minus 0.2°F., by means of the hand shutoff valve on the outgoing side of the drinking water cooler and the pressure regulating valve on the ingoing side. In order to be sure that maximum flow for these conditions is obtained with self-contained drinking water coolers, caution should be taken to see that the condensing unit is operating continuously.

4.17 For bubbler-type water coolers rated less than four (4) gallons per hour or for faucet-type water coolers, make intermittent draws into a paper cup. Use mercury-in-glass thermometer placed in water in cup to determine water temperature. Establish frequency and size of draw to result in 50°F. water being drawn. Collect, in a measuring receptacle, all water drawn during test period to check capacity.

4.18 Set the refrigerant control device, if it is adjustable, within the limits recommended by the manufacturer.

4.19 Adjust or bridge the temperature control device so that continuous operation during the test is assured.

4.20 Operate the drinking water cooler until steady temperatures and mechanical equilibrium are established and then start taking readings.

4.21 Ambient temperatures shall be maintained within plus or minus 1°F. of the specified value and shall be measured at points located three (3) feet from the floor and ten (10) feet from the sides and front of the water cooler. This condition shall be maintained during stabilization periods as well as during actual test runs.

5.10 Test Data

5.11 Identification data to be included in the test report are:

a. Nameplate data of water cooler including make, model, etc.
b. Kind of cooler, that is, either pressure bubbler with precooler, pressure bubbler with no precooler, pressure faucet or bottle faucet.
c. Motor nameplate data, including make, type, frequency, etc.

5.12 Test data to be recorded after drinking water cooler specified rating conditions have become established are:

a. Temperature of ingoing drinking water.
b. Temperature of outgoing drinking water.
c. Gallons of drinking water cooled per hour.
d. Gallons of water by-passed per hour through precooler, if provided.
e. Ambient temperature.
f. Temperature of cooling water ingoing to refrigerant condenser.
g. Temperature of cooling water outgoing from refrigerant condenser.
h. Gallons of refrigerant condenser cooling water per hour.
i. Volts at motor service connection.
j. Watts input to drinking water cooler.

5.13 All readings shall be taken every 15 minutes; except outlet drinking water temperatures during intermittent draws, which shall be taken during each draw and except inlet drinking water temperature which, in the case of the bottled water coolers, needs to be taken only at the beginning of the test and when a bottle is replaced.

5.14 The test shall be continued until eight successive readings are within the allowable limits given under "Procedure 4.10," par. 4.11 through par. 4.20.

5.15 The A. S. R. E. Standard Capacity Rating shall be an average of the eight (8) successive readings required by par. 5.14.

ASRE Circular No. 19-40

ASRE STANDARD

WATER CONTENT LIMITS FOR REFRIGERATING SYSTEM PARTS

Adopted by the Council of The American Society of Refrigerating Engineers,
June 6, 1940

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40TH ST., NEW YORK 18, N. Y.

PRICE 25 CENTS

Special A. S. R. E. Standards Subcommittee on Moisture Content

W. B. Anderson, *Chairman*
R. A. Fuller, *Secretary*
H. A. Brysselbout
W. A. Grant
R. J. Thompson

A. S. R. E. Standards Committee

Chester Lichtenberg, *Chairman*

Glenn Muffly
H. D. Edwards
L. S. Morse

L. L. Lewis
H. E. Keeler
W. R. Woolrich

Standard Water Content Limits For Refrigerating System Parts

The purpose of this standard is to establish the maxi- n permissible water content of sealed refrigerant sys- tubing and cooling coils as received at the point of allation.

The dehydration of refrigerant system tubing and cool- coils before sealing shall be such that, with proper ng with dry air or other suitable gas and sealing, these ts will not contain more than the maximum permissible er content as specified in paragraph 5.

The sealing of refrigerant system tubing and cooling s shall be such that the conditions within the tube at e of sealing shall be maintained until the seal is prop- broken.

The refrigerant system tubing and cooling coils shall filled with dry air or other suitable gas under sufficient ssure that when the seal is broken no air shall be wn into the tubing or cooling coil.

The maximum permissible water content of refriger- system tubing or cooling coils as received at the point nstallation shall be 10.6 milligrams per liter (300 milli- ms per cubic foot) of internal volume of the tubing coil.

A standard test method shall be used in determining water content of refrigerant system tubing or cooling s. The method is described in the following paragraphs.

METHOD FOR DETERMINING MOISTURE CONTENT

7 **Equipment and Chemicals.** (The letters and numerals er to Figs. 1 and 2.)

. An oven "O" in which a temperature of 215°F. to 35°F. is maintained at all points where tubing or coils o be analyzed are located.

. A vacuum pump "V" and vacuum system capable of naintaining within 1 mm. mercury of zero pressure bsolute or less.

. A manometer "M" readable to 1 mm. mercury to vhich is applied a scale graduated to 1 mm. capable of eading pressures up to 1 atmosphere absolute.

. A weighing balance sensitive and readable to 0.2 mg.

. Glass "Nesbit" type absorption bulbs, "A." The veight of these bulbs when filled with the specified ab- orbing chemical shall not be greater than the maximum pecified to obtain the sensitivity of the balance. The stoppers shall be so constructed as to prevent any leakage.

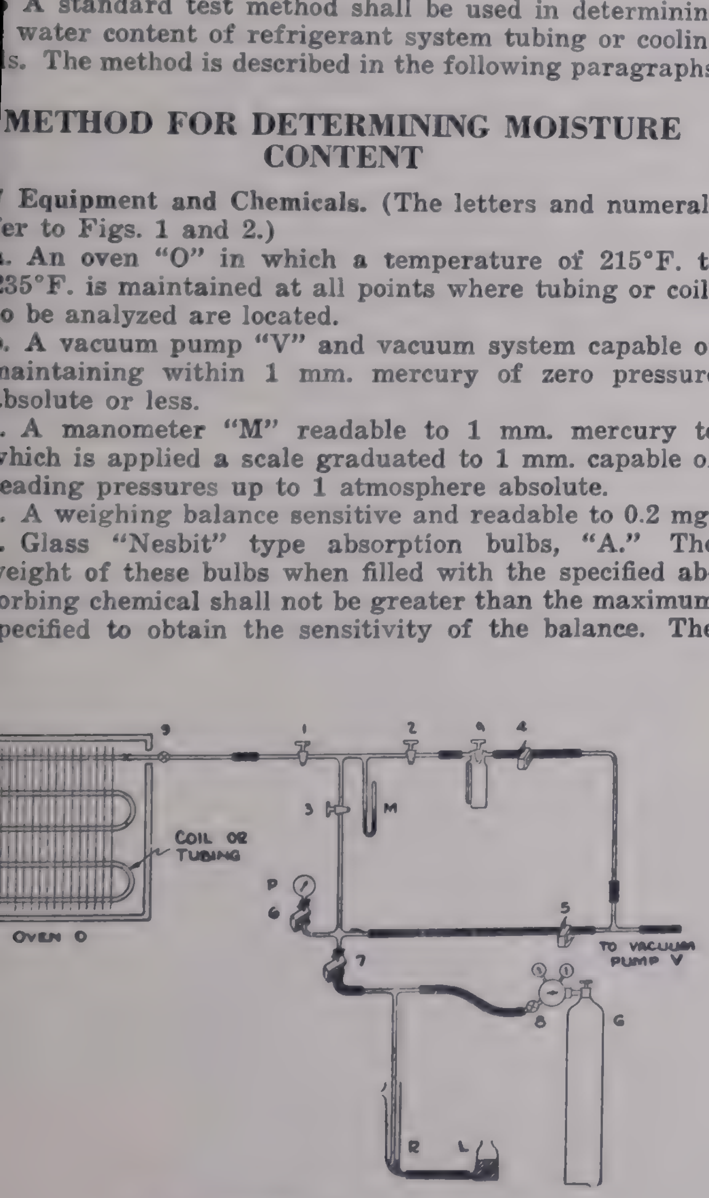


FIG. 1. DIAGRAM OF APPARATUS.

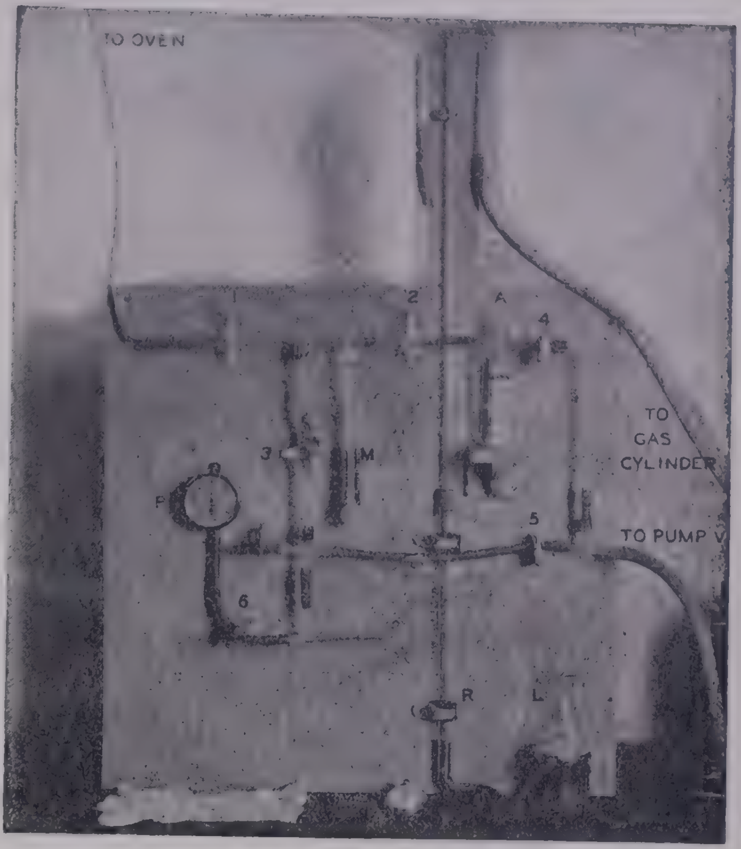


FIG. 2.

f. A pressure gage "P" sensitive to 1 mm. mercury pressure.

g. A pressure reducing valve capable of reducing gas pressure from 2000 lb. per sq. in. to 1 in. mercury gage pressure.

h. A pressure regulator "R" capable of controlling the gas pressure within 1 mm. at 30 mm. One type is shown in Fig. 3.

i. Vacuum tight glass stopcocks or vacuum tight valves, shown in Fig. 1.

j. Copper tubing, ¼ in. or 5/16 in. O.D., with suitable flared fittings to connect to vacuum-tight valve 9, and from valve to coil or tubing to be tested. All must be dried before using by heating 8 hr. at 212°F.

k. Vacuum type rubber tubing which has been dried by heating for 8 hr. at 212°F.

l. Screw clamps for rubber tubing.

m. Cylinder of compressed nitrogen, hydrogen, or air, "G", in which the pressure will at all times be above 300 lb. per sq. in. gage.

n. Phosphorous pentoxide and phosphorous pentoxide-asbestos mixture. The Nesbit type absorption bulb shall be filled with these drying chemicals in the manner shown in Fig. 4. The P_2O_5 asbestos mixture used for filling the tubes is made up by shaking equal parts, by weight, of Gooch crucible asbestos (dried at 140°C.) and P_2O_5. This mixing is best done in a closed wide mouth jar.

o. Glass wool. Asbestos.

p. Clean cotton gloves. Watch or clock.

8 Preparation of Tubing and Coils to be Tested.

a. The seals of the tubing or coils to be tested are to remain unbroken until specifically directed to open them. The tubing or coils to be tested are to be maintained at normal room ambient temperature of approximately 70°F. for a sufficient period before the test to insure that temperatures throughout are at this ambient temperature.

b. With a tubing cutter, open one of the sealed ends of the coil or tubing and quickly attach a dried vacuum tight valve 9. The connection between the coil or tubing to be tested and the vacuum-tight valve 9, shall be dry copper tubing with dry flared connections. Flared connections must be tightened with two wrenches to insure freedom from leaks. The time elapsing between opening the coil or tubing to be tested and final tightening connections should not exceed four minutes.

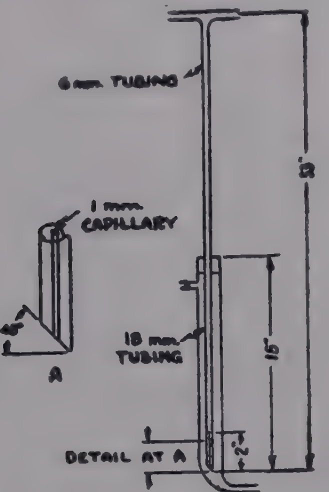


FIG. 3. PRESSURE REGULATOR.

c. Place the coil or tubing in the oven and attach a dried copper line to the vacuum-tight valve 9. The other end of this copper line is attached to the testing apparatus at valve 1. Do not open valve 9 until directed later on in the testing procedure.

d. The tubing or coil must be in the oven for sufficient time to come to the specified temperature before a test is made.

9 Procedure for Determining the Moisture Content.

a. Refer to Figs. 1 and 2 for diagram and photograph, respectively, of testing apparatus and the location of items.

b. While the oven is coming to the specified temperature, close valves 6, and "A" and open valves 1, 2, 3, 4, 5, 7. Start pump "V" and evacuate the system to a pressure of 1 mm. of mercury or less.

c. During evacuation, close valves 3 and 5 and observe manometer "M" for pressure rise. If any pressure rise is observed, correct the leak. When no pressure rise is observed proceed as follows, with valves 3 and 5 closed.

d. Close valve 2, then open valve "A" slowly to evacuate absorption bulbs "A." Then open valve 2 again.

e. Keeping valves 3 and 5 closed, open valve 8 and watch pressure regulator "R" until gas bubbles out at bottom. Set reducing valve on tank "G" to give a small but steady stream of gas bubbles.

f. Open valve 6 and adjust the mercury level in "R" using the leveling bulb "L" so that a pressure of 30 mm. ± 1 mm. is registered at gage "P". Then close valve 6.

g. Fifteen minutes after the vacuum pump was started, close valve 4. Open valve 3 carefully to allow pressure from tank "G" to build up in Nesbit bulb.

h. Open valve 6 and watch gage "P" to read the pressure obtained. When the system is at 30 mm. ± 1 mm. close valve 7 and watch gage "P" for drop-out. If any drop-out is observed, the P_2O_5 in the absorption bulb is still absorbing gas and the pressure must be re-applied. If no drop out is observed, close valve 6.

i. Close valves 2, 4 and "A", in that order. absorption bulb from its connections, handling only with clean white gloves. (The absorption bulb is never handled with bare hands).

j. Place the absorption bulb on the hanger on the balance and allow it to remain there with the door closed for exactly 10 minutes as timed by a watch or clock. The

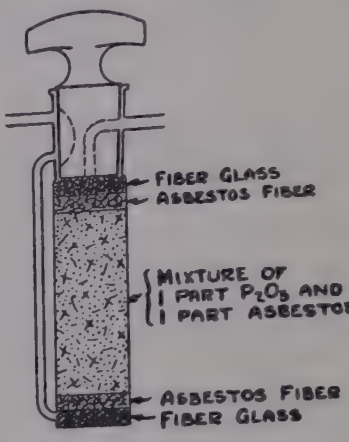


FIG. 4. ABSORPTION BULB.

absorption bulb should be counterpoised by a bulb of similar size and material.

k. At the end of exactly 10 minutes, weigh bulb to ± 0.0002 gm.

l. Using gloves, replace the absorption bulb in the testing apparatus. Open valves 5, 2, 4 and pump down system until the manometer "M" reaches the previous reading.

m. Close valve 5, open valve 7 carefully and allow pressure to build up through regulator "R".

n. Close valve 3, open valve "A" carefully and evacuate absorption bulb for 15 minutes.

o. At the end of 15 minutes, repeat the operations specified in paragraphs g, h, i, j, k, and l. The weight of the Nesbit bulb should check the previous reading within ± 0.0002 gm. If it does not, repeat the operations in paragraphs m, n, and o. When check readings are obtained proceed as follows after replacing the Nesbit and pumping down as in paragraph l.

p. Close valve 5, open valve 7 carefully and allow pressure to build up through regulator "R". Close valve 3, open valve "A" carefully and evacuate until mercury in "M" falls to its previous reading.

q. Now open valve 9 at the tubing or coil very slowly and carefully. The coil or tubing will now be evacuated through the absorption bulb. Continue pumping for one hour.

r. During this evacuation, the mercury in "M" should fall slowly to the point previously registered. If not, continue pumping until it does fall to that point.

s. Close valve 4, open valve 3 and allow pressure to build up in the entire system. Watch pressure regulator "R" for bubbles indicating that the system is full of gas. Then close valve 1 and proceed as in paragraphs h, i, j and k.

t. The gain in weight of the absorption bulb during the test is the amount of moisture contained in the coil or tubing. This weight divided by the internal volume of the coil or tubing in cubic feet shall be compared with the maximum permissible water content to determine whether dehydration methods have been satisfactory.

u. If another determination is to be made immediately afterwards, the procedure will be to replace the absorption bulb in the apparatus, then follow paragraphs 8a, b, c, and d in preparing the next coil or tubing to be tested. Open value 5, then 1 and evacuate until mercury in "M" reaches its previous low point. Continue evacuation fifteen minutes. Then proceed as in paragraphs l, m et seq.

A.S.R.E. Circular No. 20

A. S. R. E. STANDARD METHODS OF RATING AND TESTING EVAPORATIVE CONDENSERS

by

THE JOINT COMMITTEE

ON

RATING COMMERCIAL REFRIGERATING EQUIPMENT

Sponsored by The American Society of Refrigerating Engineers

APPROVED
by A.S.R.E. Council
December, 1941

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
WEST 40th STREET NEW YORK 18, N. Y.

Price 35 Cents

Joint Committee on
Rating Commercial Refrigerating Equipment

GLENN MUFFLY, Springfield, Ohio, *General Chairman*
L. S. MORSE, York, Pa., *Secretary*

For The American Society of Refrigerating Engineers (Sponsor)—
Glenn Muffly, Chairman
R. H. Tull
Frank R. Zumbro

For The Air Conditioning & Refrigerating Machinery Association—
Donald E. French
Louis S. Morse

For the American Society of Heating and Ventilating Engineers—
F. H. Faust

A.S.R.E. Standard Methods of Rating and Testing Evaporative Condensers

Sections (see last page for index)

1. Scope and Purposes
2. Classification
3. Method of Rating
4. Standard Test Methods
5. Instruments
6. Apparatus
7. Test Procedure
8. Definitions

Section 1. SCOPE AND PURPOSES

0. *Scope*

1.11. This Standard prescribes the methods of rating and testing evaporative condensers.

0. *Purposes*

1.21. The purposes of this Standard are:

a. To list and define the terms used in the rating of evaporative condensers.

b. To establish conditions and methods of test which shall be used for obtaining *A.S.R.E. Standard Ratings* of evaporative condensers.

c. To list and define the conditions which should be specified for each set of *Application Ratings* of evaporative condensers.

1.22. Application factors encountered in practice vary through such wide limits and the capacity of an evaporative condenser depends so much upon its design and the conditions under which it is installed and operated that this Standard does not attempt to include all of the installation and operation variables.

Section 2. CLASSIFICATION

0. *Classification*

2.11. Classification of evaporative condensers will be made as follows:

a. Method of water usage.

b. Type of fan.

0. *Methods of water usage are as follows:*

a. Non-recirculation of water.

b. Recirculation of water.

0. *Types of fans are as follows:*

a. Axial flow fan.

b. Radial flow fan.

c. Combination axial and radial flow fan.

Section 3. METHOD OF RATING

0. *Ratings*

3.11. Ratings of evaporative condensers may in general be considered as *A.S.R.E. Standard Ratings* or *Application Ratings* as defined and explained in paragraphs 3.12 and 3.13, respectively.

3.12. *A.S.R.E. Standard Ratings* are simplified ratings for use where a single statement or specification of performance is desired. Such ratings may be used for comparison since they are established on the basis of standardized conditions as defined in this section and selected from paragraph 3.21.

3.13. *Application Ratings* are those used in the selection and application of the product. They are usually given for a range of conditions encountered in practice so that the ratings at any desired condition may be selected or interpolated. The intent of this code as regards *Application Ratings* is to specify the terms that define the ratings and to list the conditions or basis of rating in accordance with paragraph 3.22 which should be specified and accompany such *Application Ratings*.

3.14. The capacity ratings of an evaporative condenser may be truly stated by its refrigerant heat rejection. However, an evaporative condenser is a component part of a refrigeration system, and since it is conventional practice to specify capacity ratings in terms of refrigeration effect, this method is used as a basis for this Standard.

3.15. Rating of evaporative condensers shall be expressed in the following terms:

a. The *Condensing Refrigeration Effect* in Btu. per hr.

b. The *Subcooling Refrigeration Effect* in Btu. per hr.

c. The total water supply in gal. per hr.

d. The water overflow in gal. per hr.

e. The total fan motor power input in watts or kilowatts.

f. The total pump motor power input in watts or kilowatts.

g. Air flow through the unit in cu. ft. per min. of Standard Air.

h. The value of external resistance for the rated air flow in in. of water gage.

3.16. Published ratings shall include *A.S.R.E. Standard Ratings*, identified as such.

3.20. *Standard Rating Basis*

3.21. *A.S.R.E. Standard Ratings* shall be given under conditions as specified in Table I.

Table 1.

A.S.R.E. Rating Condition	*Value*
Barometric pressure	29.92 in. mercury
Dry bulb temperature of air entering unit	90° F.
Wet bulb temperature of air entering unit	75° F.
Dry bulb temperature of ambient air	90° F.
Temperature of water entering unit	75° F.
Saturated temperature of dry refrigerant vapor entering unit	105° F.
Actual temperature of dry refrigerant vapor entering unit	Not less than 125° F.

Ratio of *Condensing Heat Rejection Effect* to the *Condensing Refrigeration Effect* shall be arbitrarily assumed as 1.20 for the above rating conditions.

3.22 Each set of *Application Ratings* should be accompanied in some manner by specifications of the following conditions:

a. Barometric pressure in in. of mercury.

b. Ambient temperature in ° F.

c. Saturated and actual temperatures of refrigerant vapor entering *Refrigeration Compressor* in ° F.

d. Saturated temperature of refrigerant vapor entering unit, in ° F.

e. Temperature of water entering unit in ° F.

f. Temperature of air entering inlet of unit in ° F. dry bulb and ° F. wet bulb.

g. Saturated and actual temperature of refrigerant liquid leaving unit, in ° F.

h. The ratio of the *Condensing Heat Rejection Effect* to the *Condensing Refrigeration Effect* for each condition of saturated and actual temperatures of the refrigerant vapor entering and leaving the *Refrigeration Compressor*.

Section 4. STANDARD TEST METHODS

4.10. *Standard Test Methods*

4.11. Standard test methods shall be used to determine the refrigerant heat rejections on which *A.S.R.E. Standard Ratings* are based.

4.12. A primary test and a simultaneous confirming test shall be used for determination of the refrigerant heat rejections under the conditions specified in paragraph 3.21.

4.13. The results of the confirming test shall be within plus or minus five (5) per cent of the primary test, but the primary test shall govern for rating purposes.

4.20. *Primary Test Methods*

4.21. The primary test shall consist of a measurement of the heat rejected from the refrigerant by determination of:

a. *Enthalpy* difference of the entering and leaving refrigerant fluid.

b. Rate of refrigerant flow.

4.22. The heat rejected from the refrigerant by the evaporative condenser is the product of the rate of refrigerant flow and the *enthalpy* difference of the entering and leaving refrigerant fluid.

4.23. *Enthalpy* difference of the entering and leaving refrigerant fluid shall be determined from the pressures and temperatures of the entering and leaving refrigerant fluid.

4.24. Rate of refrigerant flow shall be determined by use of one of the following apparatus:

a. Refrigerant Calorimeter.

b. Refrigerant Liquid Quantity Meter.

4.25. A Refrigerant Calorimeter is an apparatus that provides a means of accurately measuring the amount of heat required to evaporate and superheat the refrigerant flowing through it. The rate of refrigerant flow is given by dividing the heat input including leakage losses by the *enthalpy* difference of the entering and leaving refrigerant fluid. The calorimeter shall be selected from the types listed and described in Section 6.

4.26. A Refrigerant Liquid Quantity Meter gives the quantity in a measured period of time. The rate of flow is determined from observed data. The quantity meter shall be selected from the types listed in section 6.

4.30. *Confirming Test Methods*

4.31. The confirming test shall consist of secondary determination of the heat rejected from the refrigerant by use of one of the following apparatus:

a. Air flow measuring equipment

b. Refrigerant Liquid Quantity Meter

c. Refrigerant Liquid Flow Meter.

4.32. A Refrigerant Liquid Quantity Meter or a Refrigerant Liquid Flow Meter shall not be used for the confirming test if a Refrigerant Liquid Quantity Meter is used for the primary test.

4.40. *Confirming Test with Air Flow Measuring Apparatus*

4.41. When air flow measuring apparatus is used the confirming test shall provide determination of heat rejected from the refrigerant in the evaporative condenser from the following data:

a. Air flow rate.

b. *Enthalpy* of entering and of leaving air.

c. Water flow rate in, and water flow rate out.

d. *Enthalpy* of entering and leaving water.

e. Net heat input of fan and pump motors.

f. Heat leakage losses.

4.42. The basis of the determination is the heat balance equation:

$$q_t = q_a + q_w + q_{ho} - q_{fm} - q_{pm} - q_{hi}$$

where q_t = Heat added by the refrigerant fluid

q_a = Heat removed by air stream, in Btu. per hr.

q_w = Heat removed by water stream

q_{ho} = Heat removed by heat leakage through external surfaces of evaporative condenser

q_{fm} = Heat added by the fan motor

4.42.

q_{pm} = Heat added by the pump motor

q_{hi} = Heat added by heat leakage through external surfaces of the evaporative condenser.

4.43. Air flow rate shall be determined by the use of either the Receiving Chamber with Impact Nozzle or the Receiving Chamber with Static Nozzle, as listed and described in Section 6.

4.44. *Enthalpy* of entering and of leaving air shall be determined from the entering and leaving dry bulb and wet bulb temperatures, and the barometric pressure.

4.45. Water flow rate in and out shall be determined by use of water quantity or flow meters.

4.46. The net heat input of the fan and pump motors shall be determined from motor inputs as measured by a watt meter or watt hour meter. The net heat input of motors and drives not in the air stream shall be obtained by multiplying the observed watts input reading by the motor and drive efficiency. Fan motor heat inputs are subject to a further correction for the kinetic energy of the leaving air stream.

4.47. Heat leakage losses from the external surfaces of the evaporative condenser above or below the ambient temperature shall be minimized by insulation with one-half (0.5) in. thickness of cork or its equivalent, and no further correction shall be necessary for this effect. When it may be shown that without such insulation the net heat leakage through the external surfaces of the evaporative condenser, $q_{hi} - q_{ho}$, which may be either positive or negative, is not more than one (1) per cent of heat rejected from the refrigerant fluid, such insulation may be omitted.

. *Confirming Test Methods with Refrigerant Liquid Quantity or Flow Meter*

4.51. When a refrigerant liquid quantity or flow meter is used, the determination of refrigerant heat rejected from the evaporative condenser shall be made by determination of:

a. Enthalpy difference of entering and leaving refrigerant fluid.

b. Rate of refrigerant flow.

4.52. *Enthalpy* difference between the entering and leaving refrigerant fluid shall be determined from the pressures and temperatures of the refrigerant fluid at the points where it enters and leaves the evaporative condenser.

4.53. The Refrigerant Liquid Quantity Meter gives the quantity in a measured period of time. The rate of flow is determined from observed data. Selection of this apparatus shall be made from types described in Section 6.

4.54. A Refrigerant Liquid Flow Meter indicates the instantaneous rate of flow. Selection of this apparatus shall be made from the types described in Section 6.

4.55. Air flow through the unit in cubic feet per minute of *Standard Air* shall be determined by use of the Duct and Pitot Tube apparatus as listed and described in Section 6.

Section 5. INSTRUMENTS

5.10. *Selection*

5.11. Selection of instruments for rating tests shall be made from the types listed in this section.

5.20. *Accuracy*

5.21. Accuracy of instruments used for rating tests shall be within the limits specified in this section.

5.30. *Temperature Measuring Instruments*

5.31. Temperature measurements shall be made with one or more instruments as follows:

a. Mercury-in-glass thermometers

b. Thermocouples

c. Electric resistance thermometers.

5.32. Accuracy of temperature measuring instruments shall be within limits as follows:

a. Wet and dry bulb air temperatures, ± one-tenth (0.1) °F.

b. Refrigerant temperatures, ± one-tenth (0.1) °F.

c. Temperatures of brine or water in calorimeter, ± one-tenth (0.1) °F.

d. All other temperatures, ± five-tenths (0.5) °F.

5.40. *Pressure Measuring Instruments*

5.41. Pressure measurements shall be made with one or more instruments as follows:

a. Mercury column.

b. Bourdon Tube gage.

5.42. Accuracy of pressure measuring instruments shall permit pressure measurements within the limits of ± one (1.0) per cent absolute.

5.50. *Air Capacity Measuring Instruments*

5.51. Air capacity measurements shall be with a manometer used in conjunction with either a nozzle or a combined (double tube) pitot tube.

5.52. The nozzle shall be as specified in the latest approved edition of the A.S.M.E. Power Test Codes, Instruments and Apparatus, Part IV. Measurement of Quantity of Materials, Chapter 4. Flow Measurement by Means of Standardized Nozzles and Orifice Plates.

5.53. The combined (double tube) pitot tube shall conform to the following specifications:

a. The tip of the pitot tube shall be either conical or hemispherical in shape.

b. The diameter of the impact opening shall not be less than two-tenths (0.2) nor greater than five-tenths (0.5) of the outside diameter of the pitot tube.

c. The static orifices shall be located at a distance not less than eight (8) tube diameters from the point where the straight tubing joins the pitot tube tip.

d. The diameter of the static orifices shall be not less than two one-hundredths (0.02) in., nor greater than four one-hundredths (0.04) in.

e. There shall not be less than four static orifices.

5.58.
- f. The pitot tube stem shall be located at least eight (8.0) diameters of straight tubing from the static orifices farthest from the pitot tube tip.
- g. The outer tube shall be smooth and polished; the inner tube concentric with the outer tube, and the static orifices shall be free of burrs and irregularities.

5.54. Accuracy shall be as follows:
- a. The manometer shall be capable of being read to within five one-thousandths (0.005) in. water gage.
- b. The nozzle coefficient shall be determined within plus or minus one (1.0) per cent.

5.60. *Electrical Measuring Instruments*

5.61. Electrical measurements shall be made with one or more of the following instruments:
- a. Indicating
- b. Integrating

5.62. Accuracy of electrical measuring instruments shall be within limits as follows:
- a. Indicating instruments within ± five-tenths (0.5) of one per cent.
- b. Integrating instruments within ± one (1.0) per cent.

5.70. *Water Flow Measuring Instruments*

5.71. Water flow measurements shall be made with one or more instruments as follows:
- a. Liquid quantity meter, measuring either weight or volume.
- b. Liquid flow meter.

5.72. Accuracy of water flow measuring instruments shall be within plus or minus one (1.0) per cent.

5.80. *Speed Measuring Instruments*

5.81. Speed measurements shall be made with one or more instruments as follows:
- a. Revolution counter
- b. Tachometer
- c. Stroboscope
- d. Oscillograph.

5.82. Accuracy of speed measuring instruments shall be within ± one-quarter of one (0.25) per cent.

5.90. *Measurements*

5.91. Time measurements shall be made with instruments whose accuracy is within plus or minus one-tenth of one (0.1) per cent.

5.92. Weight measurements shall be made with instruments whose accuracy is within ± two-tenths of one (0.2) per cent.

Section 6. APPARATUS

6.10. *Selection*

6.11. Selection of apparatus for rating tests shall be made from those listed and described in this section.

6.20. *Calorimeters for Measuring Rate of Heat Flow*

6.21. The refrigeration calorimeter shall be selected from the following types:
- a. Brine Tank Calorimeter
- b. Concentric Tube Calorimeter
- c. Secondary Refrigerant Calorimeter
- d. Flooded or Dry System Primary Refrigerant Calorimeter.

6.22. A Brine Tank Calorimeter includes an evaporator and a heating unit submerged in an externally insulated brine bath. The refrigerating effect produced in the evaporator by operation of a compressor and evaporative condenser is balanced by the heat added to the brine tank.

6.23. A Concentric Tube Calorimeter includes a heat exchanger for the transfer of heat from a fluid heating medium to a vaporizing refrigerant. The refrigeration effect produced by the operation of the compressor and evaporative condenser is balanced by heat given up by the fluid heating medium.

6.24. A Secondary Refrigerant Calorimeter includes an evaporator located in the top and a heating unit located in the bottom of a pressure tight container which is partially filled with a secondary refrigerant. The refrigeration effect produced in the evaporator by the operation of the compressor and evaporative condenser is balanced by heat added to the secondary refrigerant.

6.25. A Flooded or Dry System Primary Refrigerant Calorimeter includes an evaporator with a heater in direct thermal contact with the primary refrigerant. The refrigerating effect produced through operation of the compressor and evaporative condenser is balanced by heat added to the primary refrigerant.

6.26. The refrigerant calorimeters shall be constructed so as to conform to standards specified in the latest approved edition of the Standard Methods of Rating and Testing Mechanical Condensing Units of The American Society of Refrigerating Engineers.

6.27. Calibration of the refrigerant calorimeters shall be made in accordance to methods specified in the latest approved edition of the Standard Methods of Rating and Testing Mechanical Condensing Units of The American Society of Refrigerating Engineers.

6.28. The refrigerant calorimeters shall be operated so that the entering refrigerant liquid shall have a temperature at least five (5.0) °F. lower than its saturated temperature, and the leaving refrigerant vapor shall have at least ten (10.0) °F. superheat. If steam is the heating medium, the entering steam shall have at least (8.0) °F. superheat, and the condensate shall either be subcooled or the collecting vessel shall be adequately covered.

6.30. *Quantity and Flow Meters for Determining Rate of Refrigerant Flow*

6.31. Refrigerant meters shall be selected from the following types:
- a. Refrigerant Liquid Quantity Meter

6.31.

b. Refrigerant Liquid Flow Meter.

6.32. A Refrigerant Liquid Quantity Meter is an apparatus that measures the circulation of the refrigerant. It includes hand operated measuring cylinders or an integrating mechanism and supplementary devices for determining the quantity of refrigerant liquid circulated in a given period of time. Where a Refrigerant Liquid Quantity Meter is used for determination of the primary ratings it shall be of the hand operated type.

6.33. A Refrigerant Liquid Flow Meter includes a mechanism and supplementary devices for determining the instantaneous rate of circulation of refrigerant liquid.

6.34. Refrigerant liquid quantity and flow meters shall be constructed and equipped so as to conform to the standard specified in the latest approved edition of the Standard Methods of Rating and Testing Mechanical Condensing Units of The American Society of Refrigerating Engineers.

6.35. Calibration of refrigerant liquid quantity and flow meters shall be performed in accordance to the methods specified for such instruments in the Standard Methods of Rating and Testing Mechanical Condensing Units of The American Society of Refrigerating Engineers.

6.36. Accuracy of liquid refrigerant quantity and flow meters shall be within ± one (1.0) per cent.

6.37. At the time of test, a sample of refrigerant shall be taken to determine the amount of oil in circulation.

6.38. Refrigerant liquid quantity and flow meters having metering or other mechanisms causing pressure drop in or through the meter shall be provided with a means for supplying the refrigerant liquid with sufficient subcooling to prevent vaporization of the liquid in the meter. Sight glasses shall be provided to determine that no vapor bubbles are mixed with the refrigerant liquid.

0. *Air Measuring Apparatus*

6.41. Air measuring apparatus shall be selected from the following types:

a. Receiving Chamber with Impact Nozzle
b. Receiving Chamber with Static Nozzle
c. Duct and Pitot Tube.

0. *Receiving Chamber with Impact Nozzle*

6.51. The Receiving Chamber with Impact Nozzle measures the velocity pressure and the dry bulb and wet bulb temperatures at the throat of a calibrated nozzle placed between a receiving and a discharge chamber.

6.52. A chamber is constructed for receiving and mixing the air discharged from the unit. It shall be well insulated with two (2.0) in. of cork or equivalent and sealed, if possible, so that air leaking at any static pressure within the range of the tests shall not exceed one (1.0) per cent of the total air flow. In no case shall the leakage be greater than five (5) per cent. If leakage is over one (1.0) per cent, the chamber shall be calibrated for leakage at various static pressures covering the test range. The chamber shall be of such a size that the unit to be tested will produce from twenty (20) to ninety (90) air changes per minute.

6.53. The outlet or outlets of the unit to be tested shall be joined directly to the receiving chamber with a minimum of communicating duct.

6.54. Two or more static orifices shall be located in the receiving chamber and not in the direct air blast from the unit. These openings shall be connected by air-tight tubes to a common manometer (draft gage) which can be read to within five one-thousandths (0.005) in. of water.

6.55. A calibrated nozzle shall be fitted into the wall of the chamber, discharging into the duct leading to the exhaust fan. The outlet opening of the nozzle shall be of such area that the air velocity is not less than three thousand (3000) ft. per min.

6.56. An exhaust fan shall be connected to the discharge chamber and shall be of such capacity that it will overcome the resistance of the receiving chamber, the discharge duct, and the nozzle. It shall be able to provide a zero static pressure at the outlet of free delivery units.

6.57. Two manometers (draft gages) in parallel shall be provided for measuring pressures at the nozzle. One side of each manometer shall be connected to a static orifice located flush with the inner wall of the exhaust duct and near the chamber wall. The other side of each manometer shall be connected to an impact tube arranged to measure the velocity pressure of the air in the nozzle outlet.

6.58. Wet bulb and dry bulb temperature measurements shall be made with temperature measing instruments located in the throat of the nozzle. At least one pair of instruments shall be used for each square foot of throat area.

6.60. *Receiving Chamber with Static Nozzle*

6.61. The Receiving Chamber with Static Nozzle method is substantially the same as that of the Receiving Chamber Method with Impact Nozzle except that the receiving chamber is much larger so that the velocity pressure of the air approaching the nozzle is reduced to a negligible amount and consequently only static pressure drop need be measured.

6.62. The chamber for receiving and mixing the air discharge from the unit shall be constructed as described in paragraph 6.52 except that it shall be of such size that the unit to be tested will produce from one (1) to ten (10) air changes a minute and it need not be insulated. If desired, a well gasketed door may be provided for entering the chamber. The area of the wall of the receiving chamber, perpendicular to the axis of the air flow from the unit being tested, shall be such that the average air velocity shall not exceed one hundred and fifty (150) ft. per min. as calculated from the total air flow divided by the said wall area.

6.63. The outlet or outlets of the unit to be tested shall be joined directly to a mixing box in which temperature measurements are made in restricted openings or nozzles. This mixing box shall connect directly to one wall of the receiving chamber without a communicating duct, and shall contain one or more static orifices not in the direct blast from the unit. These static orifices shall be connected by means of air-tight tubes to a common manometer (draft gage).

6.64. One or more calibrated nozzles shall be fitted into one wall of the receiving chamber and discharge into a discharge chamber. The nozzle velocity shall be not less than two thousand (2000) ft. per min. Multiple nozzles in parallel are recommended for high values of air quantity. Center-to-center distances between nozzles in use shall not be less than three (3) throat diameters and the distance from the center line of any nozzle to any of the four adjacent side walls shall not be less than three (3) throat diameters. If the nozzles are of different diameters the distance between axes shall not be less than three (3) times the diameter of the larger throat.

6.65. The possibility of any of the discharge air entering directly into the nozzle shall be minimized by locating the nozzle in a wall adjacent to the wall where the air enters the chamber, and at a different level. The average approach velocity shall not be greater than three (3) per cent of the nozzle velocity and the maximum approach peak velocity shall not exceed (6) per cent of the nozzle velocity. The approach velocity shall be defined as the average velocity of the air in the receiving chamber, as calculated by dividing the total air flow by the area of the wall of the receiving chamber (in which the nozzle is located), which is perpendicular to the axis of the nozzle.

6.66. The size and arrangement of the discharge chamber shall be such that the effect of air velocities on the pressure readings on the discharge side of the nozzle shall not exceed one (1) per cent. The distance from the nozzle to the nearest obstruction shall not be less than ten (10) throat diameters, unless suitable diffusion plates or baffles are used. The area of the wall on the discharge side of the nozzle shall be at least one-half (0.5) the area of the chamber side of the same wall.

6.67. An exhaust fan shall be connected to the discharge chamber and shall be of such capacity that it will overcome the resistance of the chambers, nozzles, and mixing box. It shall be able to provide a zero static pressure at the outlet of free delivery units. Means shall be provided with which to vary the capacity of the exhaust fan.

6.68. Two or more manometers (draft gages) in parallel, having one side connected to a static orifice located flush with the inner wall of the receiving chamber, and the other side connected in a similar manner to a static orifice in the wall of the nozzle discharge chamber, shall be used to obtain the pressure drop across the nozzle. The static orifice shall be located so as not to be affected by the air flow.

6.69. Wet bulb and dry bulb temperature measurements shall be made with temperature measuring instruments located as follows:

a. In the outlet of the mixing box.
b. In the throat of the air measuring nozzle
c. Instruments at location (a) shall be used for measuring test discharge air temperatures instruments at location (b) shall be used for calibrating the nozzle.

6.70. *Duct and Pitot Tube*

6.71. The Duct and Pitot Tube apparatus provides a determination of the air flow rate in cubic feet per minute of *Standard Air* through the evaporative condenser by measurement of velocity pressure and air temperatures in a duct attached to the air discharge opening of the unit.

6.72. The test duct shall have an area equal to the area of the discharge opening of the unit. This duct shall be straight, of uniform area and section, and not less than ten (10) diameters in length. For rectangular ducts the diameter shall be considered as equal to the average of the dimensions of the two sides. If a transformation piece is used to join the test duct to the unit, the greatest angle between the axis of the test duct and any longitudinal element shall not exceed seven (7) degrees. Units having more than one air discharge opening shall be provided with an appropriate number of test ducts.

6.73. When it is necessary to operate the evaporative condenser at other than free delivery, the desired value of external resistance shall be obtained by the use of a symmetrical throttling device attached to the discharge opening of the test duct.

6.74. Velocity and static pressure measurements shall be made by use of a combined (double tube) pitot tube and two manometers (draft gages). Velocity pressure readings will be indicated by the manometer connected between the impact pressure and static pressure taps of the pitot tube. Static pressure readings will be indicated by the manometer with one side connected to the static pressure tap of the pitot tube, with the other side open to the atmosphere. Readings shall be taken at a location three-quarters of the length of the test duct from the air discharge opening of the unit. These readings shall be taken over a plane at right angles to the axis of the duct with the pitot tube parallel to this axis and the impact of the tube pointed upstream.

6.75. In all round ducts, not less than twenty (20) readings, consisting of not less than two (2) traverses of ten (10) readings each at right angles to each other, shall be taken at the centers of areas of five equal area concentric zones across the section of the duct.

6.76. In all rectangular ducts, the readings shall be taken in the center of equal areas over the

6.76. cross section of the duct. The number of equal areas shall not be less than sixteen (16), and need not be more than sixty-four (64). When less than sixty-four (64) readings are taken, the number of equal spaces shall be such that the centers of these rectangular areas are not more than six (6) inches apart.

6.77. The average velocity pressure of the air in the discharge duct shall be taken as the square of the arithmetic average of the square roots of the individual velocity pressure readings.

6.78. Wet bulb and dry bulb temperature measurements shall be made with temperature measing instruments located in the test duct at the discharge end. At least one pair of instruments shall be used for each square foot of duct area.

Section 7. TEST PROCEDURE

0. *Apparatus Assembly*

711. The evaporative condenser and a refrigeration compressor with an associated motor drive are connected with an appropriate combination of a refrigerant calorimeter or an evaporator and a refrigerant liquid quantity or flow meter to form a complete refrigeration system. Liquid subcoolers or refrigerant vapor superheaters may be used, if necessary, to obtain rating conditions. Provision should be made for supplying the refrigerant calorimeter with a source of heat, either steam, heated liquid, or electricity. The evaporative condenser shall be located in a room in which the dry bulb and wet bulb air temperatures can be accurately controlled. The refrigerant calorimeter shall be located in a room in which the dry bulb air temperature can be controlled.

7.12. The air flow measuring apparatus is connected to the evaporative condenser air discharge opening. A duct used to join the evaporating condenser to a Receiving Chamber with Impact Nozzle or a Receiving Chamber with Static Nozzle shall be as short as possible. Such a joining duct and all portions of the refrigerant circuits such as the liquid receiver and piping, that are located outside the unit proper, shall be insulated with one-half (0.5) in. thickness of cork or equivalent insulation to minimize heat leakage. The evaporative condenser and joining duct shall be effectively sealed against air leakage.

20. *Location and Use of Measuring Instruments*

7.21. Refrigerant temperature measuring instruments shall be located at the following points:
  a. The refrigerant vapor inlet of the evaporative condenser.
  b. The refrigerant liquid outlet of the evaporative condenser.
  c. The refrigerant liquid inlet of the refrigerant calorimeter.
  d. The refrigerant liquid inlet of the quantity or flow meter.
  e. The refrigerant vapor outlet of the refrigerant calorimeter.
  f. The refrigerant liquid outlet of the quantity or flow meter.

7.22. Refrigerant pressure measuring instruments shall be located at the following points:
  a. The refrigerant vapor inlet of the evaporative condenser.
  b. The refrigerant liquid outlet of the evaporative condenser.
  c. The refrigerant liquid inlet of the refrigerant calorimeter.
  d. The refrigerant liquid inlet of the quantity or flow meter.
  e. The refrigerant vapor outlet of the refrigerant calorimeter.
  f. The refrigerant liquid outlet of the quantity or flow meter.

7.23. Dry bulb and wet bulb air temperature measuring instruments shall be paired and shall be placed at centers of equal areas of one square foot when the air velocity over the measuring instruments exceeds four hundred (400) ft. per min. For velocities less than four hundred (400) ft. per min., the wet bulb temperature shall be obtained by use of a sling or aspirating psychrometer. All instruments shall be calibrated and shall be shielded if exposed to radiant heat. The instruments shall be located at the following points:
  a. The air inlet opening of the evaporative condenser.
  b. The throat of the nozzle of the Receiving Chamber with Impact Nozzle, or the outlet of the mixing box of the Receiving Chamber with Static Nozzle, when air flow measuring apparatus is used for the confirming test.
  c. The air discharge opening of the test duct when the Duct and Pitot Tube method of air measurement is used.

7.24. Dry bulb air temperature measuring instruments shall be placed at the following points:
  a. One or more locations to determine the average ambient temperature for the refrigerant calorimeter.

7.25. Temperature measuring instruments shall be placed at the following points:
  a. The water inlet connection of the evaporative condenser.
  b. The water outlet connection, if any, of the evaporative condenser.
  c. The steam or other heating fluid inlet of the refrigerant calorimeter, if used.
  d. The condensate or other heating fluid outlet of the refrigerant calorimeter, if used.

7.26. Air pressure measuring instruments of the manometer (draft gage) type shall be provided at the following points:
  a. The mixing box of the Receiving Chamber with Static Nozzle, or the receiving chamber proper of the Receiving Chamber with Impact Nozzle, or the static pressure connection of the combined pitot tube of the Duct and Pitot Tube apparatus.
  b. Between the receiving chamber and discharge chamber of the Receiving Chamber with Static Nozzle, or between the impact pitot tube and discharge chamber of the

7.26. Receiving Chamber with Impact Nozzle, or between the impact pressure and static pressure connections of the combined pitot tube of the Duct and Pitot Tube apparatus.

7.27. Pressure measuring instruments shall be located at the following point:
- a. The steam inlet to the refrigerant calorimeter, if steam is the heating medium.

7.28. Liquid quantity or flow measuring instruments shall be placed at the following points:
- a. The inlet water connection of the evaporative condenser.
- b. The outlet (overflow) water connection of the evaporative condenser.
- c. The outlet of the refrigerant calorimeter, when steam or other fluid is the heating medium.

7.29. Electrical measuring instruments shall be provided at the following points:
- a. In the evaporative condenser fan motor leads to measure voltage and power input.
- b. In the evaporative condenser pump motor leads to measure voltage and power input.
- c. In the refrigerant calorimeter electrical heater leads to measure power input.
- d. In the refrigerant calorimeter stirrer motor leads to measure voltage and power input.

7.30. *Preparation*

7.31. Assemble the apparatus and instruments as specified in sub-sections 7.10 and 7.20 and make required refrigerant, steam, water, electrical, etc. connections.

7.32. Place the correct oil and refrigerant charges in the proper locations in the refrigeration system.

7.33. Maintain the refrigerant at normal operating level in the evaporative condenser liquid receiver.

7.34. Check refrigeration system for vapor tightness, belts for freeness, fan, compressor and pump motors for correct rotation.

7.35. Purge the refrigeration system of non-condensible gases until the saturation temperature corresponding to the observed pressure is within one (1.0) °F. of the observed liquid temperature.

7.40. *Operation and Limits*

7.41. Start the evaporative condenser and associated apparatus and regulate and maintain the voltage at the electrical terminals of the evaporative condenser fan and pump motors to within plus or minus two (2.0) per cent of rated motor voltage. Terminal voltage and power input to the fan motor, and to the pump motor, shall be recorded at the beginning and at the end of each test.

7.42. Obtain and maintain air conditions specified in Table 1, and in accordance with the tolerances and instructions as follows:
- a. The arithmetic average of all dry bulb temperatures during the test shall not vary from the specified dry bulb temperature b more than two (2.0) °F.
- b. The arithmetic average of all wet bulb temperatures shall not vary from the specifie wet bulb temperature by more than one hal of one (0.5) °F.
- c. Individual dry bulb temperature reading shall not vary more than three (3.0) °F from that specified, and individual wet bul temperatures shall not vary more than on (1.0) °F. from that specified.
- d. All wet bulb instruments must be in *evaporative equilibrium* when read.
- e. Record all readings of inlet and outlet ai to one tenth of one (0.1) °F.

7.43. Obtain and maintain refrigerant condition specified in Table 1, and in accordance with the tolerances and instructions as follows:
- a. The arithmetic average of all pressure readings during the test shall not vary from the specified pressure by more than two (2.0) lb. per sq. in.
- b. The arithmetic average of all temperature readings during the test shall not vary from the specified temperature by more than five (5.0) °F.
- c. Record all temperature readings to one tenth of one (0.1) °F.

7.44. Obtain and maintain water conditions as specified in Table 1, and in accordance with the tolerances and instructions as follows:
- a. The arithmetic average of all temperature readings during the test shall not vary from the specified temperature by more than two (2.0) °F.
- b. The water control shall be adjusted to provide the normal water operating height or pressure.
- c. The bleeder or waste water rate shall be maintained at the normal setting.

7.45. When the Receiving Chamber with Impact Nozzle or the Receiving Chamber with Static Nozzle is used, the exhaust fan shall be controlled so that the static pressure, as indicated by the static manometer (draft gage), is maintained at zero, excepting that for pressure type units, the static pressure in the chamber at the discharge opening of the unit shall be maintained equal to the stated external resistance. When the Duct and Pitot Tube apparatus is used with pressure type units, the static pressure in the test duct shall be maintained equal to the stated external resistance by adjustment of a symmetrical throttling device attached to the discharge opening of the test duct.

7.46. Manometer (draft gage) readings of the velocity pressure with the Receiving Chamber with Impact Nozzle Method or nozzle pressure drop with the Receiving Chamber with Static Nozzle Method shall be made each time a general reading is taken and shall be immediately preceded or immediately followed by a reading of the entering and leaving dry bulb and wet

7.46. bulb temperatures. Manometer readings of the velocity pressure with the Duct and Pitot Tube apparatus shall be made at the beginning and at the end of the test. Manometer readings shall be recorded to within five one-thousandths (0.005) in. of water gage.

7.47. All readings, except for specified exceptions, shall be taken every twenty minutes and the tests shall be continued until four successive readings are within the specified limits.

7.48. The barometric pressure shall be read once during the test.

Data To Be Recorded

Data to be recorded for each rating test should include:

a. Test number
b. Date
c. Observers' names
d. Manufacturer's name and address
e. Unit size or number
f. Unit type
g. Data on each motor, as follows: (1) where used, (2) rated horse power (3) rated r.p.m. (4) manufacturer's name and address (5) model or type designation (6) rated voltage (7) phase (8) cycles.
h. Fan speed in r.p.m.
i. Pump speed in r.p.m.
j. Refrigerant used
k. Evaporative Condenser rating conditions data, as follows:
    1. Barometric pressure, in inches of mercury. Symbol = p_{ab}
    2. Dry bulb temperature of entering air, in °F. Symbol = t_{ae}
    3. Wet bulb temperature of air entering unit in °F. Symbol = t'_{ae}
    4. Dry bulb temperature of ambient air, in °F.
    5. Temperature of entering water, in °F. Symbol = t_{we}
    6. Entering refrigerant vapor pressure, in lb. per sq. in. absolute. Symbol = p_{se}
    7. Entering refrigerant vapor temperature, in °F. Symbol = t_{se}
l. Evaporative Condenser data for rating purposes, as follows:
    1. Entering water flow rate, in lb. per hr. Symbol = w_{we}
    2. Overflow water flow rate, in lb. per hr. Symbol = w_{wd}
    3. Input to fan motor, in watts. Symbol = P_{fm}
    4. Voltage at fan motor terminals.
    5. Input to pump motor, in watts. Symbol = P_{pm}
    6. Voltage at pump motor terminals.
    7. External resistance, if any, at the air discharge opening, in in. of water gage. Symbol = p_{as}
    8. Leaving refrigerant liquid pressure, in lb. per sq. in. absolute. Symbol = p_{lo}
    9. Leaving refrigerant liquid temperature, in °F. Symbol = t_{lo}
m. Refrigerant Calorimeter data, as follows:
    1. Entering refrigerant liquid pressure, in lb. per sq. in. absolute. Symbol = p_{le}
    2. Entering refrigerant liquid temperature, in °F. Symbol = t_{le}
    3. Leaving refrigerant vapor pressure, in lb. per sq. in. absolute. Symbol = p_{1e}
    4. Leaving refrigerant vapor temperature, in °F. Symbol = t_{1e}
    5. Calorimeter ambient temperature, in °F.
    6. Total electrical heat input to calorimeter in watts.
    7. Entering steam pressure in lb. per sq. in. absolute.
    8. Entering steam (or liquid) temperature in °F.
    9. Leaving condensate (or liquid) temperature in °F.
    10. Calorimeter heating fluid flow rate in lb. per hr.
    11. Stirrer motor electrical input, in watts.
    12. Calorimeter heat leakage coefficient, in Btu. per hr. per °F.
    13. Stirrer motor heat conversion factor.
    14. Total heat input to calorimeter, in Btu. per hr. Symbol = q_e
n. Refrigerant Liquid Quantity Meter data, as follows:
    1. Entering refrigerant liquid pressure, in lb. per sq. in. absolute. Symbol = p_{lme}
    2. Entering refrigerant liquid temperature, in °F. Symbol = t_{lme}
    3. Leaving refrigerant liquid pressure, in lb. per sq. in. absolute. Symbol = p_{lmd}
    4. Leaving refrigerant liquid temperature, in °F. Symbol = t_{lmd}
    5. Volumetric rate of refrigerant liquid flow, corrected for oil content, in cu. ft. per hr. Symbol = Q_{rm}
o. Liquid Refrigerant Flow Meter data, as follows:
    1. Entering refrigerant liquid pressure, in lb. per sq. in. absolute. Symbol = p_{lme}
    2. Entering refrigerant liquid temperature, in °F. Symbol = t_{lme}
    3. Leaving refrigerant liquid pressure, in lb. per sq. in. absolute. Symbol = p_{lmd}
    4. Leaving refrigerant liquid temperature, in °F. Symbol = t_{lmd}
    5. Volumetric rate of refrigerant liquid flow, corrected for oil content, in cu. ft. per hr. Symbol = Q_{rm}
p. Air Flow Measurement Method data, as follows:

7.50.

1. Dry bulb temperature of discharge air, in °F. Symbol = t_{ad}
2. Wet bulb temperature of discharge air, in °F. Symbol = t'_{ad}
3. Velocity pressure at discharge of calibrated nozzle in receiving chamber, static pressure difference across calibrated nozzle, or velocity pressure in test duct, depending upon method of air flow measurement used, in in. of water gage. Symbol = p_{av}
4. Nozzle or duct area, in sq. ft. Symbol = A
5. Nozzle discharge coefficient. Symbol = c_d
6. Temperature of overflow water, in °F. Symbol = t_{wd}
7. Fan motor and drive efficiency. Symbol = e_{fm}
8. Pump motor and drive efficiency. Symbol = e_{pm}

7.60. *Computation of Results*

7.61. Symbols used in computation of results include:

a. *Saturated Refrigerant Temperatures:*
   1. t'_{sc} = Saturated temperature of refrigerant vapor entering evaporative condenser, in °F.
   2. t'_{lc} = Saturated temperature of refrigerant liquid leaving evaporative condenser, in °F.

b. *Enthalpies*
   1. h_{sc} = enthalpy of refrigerant vapor at t_{sc} & p_{sc}, in Btu. per lb.
   2. h'_{lc} = enthalpy of a refrigerant liquid at t'_{sc} & p_{sc}, in Btu. per lb.
   3. h_{lc} = enthalpy of refrigerant liquid at t_{lc} & p_{lc}, in Btu. per lb.
   4. h_{le} = enthalpy of refrigerant liquid at t_{le} & p_{le}, in Btu. per lb.
   5. h_{1e} = enthalpy of refrigerant vapor at t_{1e} & p_{1e}, in Btu. per lb.
   6. Σ_{ae} = *Sigma Function* of air at t'_{ae} & p_{ab}, in Btu per lb.
   7. Σ_{ad} = *Sigma Function* of air at t'_{ad} & p_{ab}, in Btu. per lb.

c. *Moisture Content of Air*
   1. C_{ae} = moisture content of air at t_{ae}, t'_{ae} & p_{ab}, in lb. per lb. of dry air.
   2. C_{ad} = moisture content of air at t_{ad}, t'_{ad}, and p_{ab}, in lb. per lb. of dry air.

d. *Specific Volumes*
   1. V_{rm} = specific volume of refrigerant liquid at quantity or flow meter = cu. ft. per lb. at $\dfrac{t_{lme} + t_{lmd}}{2}$
   2. v'_{ad} = specific volume of air and water mixture at t_{ad}, t'_{ad} & p_{ab}, in cu. ft. of mixture per lb. of dry air.

e. *Flow Rates*
   1. w_r = rate of refrigerant flow in lb. per hr.
   2. w'_a = air flow rate, in lb. per min. of dry air.
   3. Q_{ad} = volumetric rate of air flow at t_{ad}, t'_{ad} and p_{ab}, in cu. ft. per min. of air and water vapor mixture.
   4. Q'_{ad} = volumetric rate of air flow at t_{ad}, t'_{ad} & p_{ab}, in cu. ft. per min. of dry air.

f. *Auxiliary Heat Flow Rates*
   1. q_a = heat removed by air stream, in Btu per hr.
   2. q_w = heat removed by discharge water in Btu. per hr.
   3. q_{fm} = heat added by fan motor, in Btu per hr.
   4. q_{pm} = heat added by pump motor, in Btu per hr.

g. *Rating Heat Flow Rates*
   1. q_t = *Total Refrigerant Heat Rejection Effect* of evaporative condenser, in Btu. per hr.
   2. q_c = *Condensing Heat Rejection Effect* of evaporative condenser, Btu. per hr.
   3. q_s = *Subcooling Refrigerant Heat Rejection Effect* of the evaporative condenser in Btu. per hr.

   Subscript *c* applied to the above symbol indicates confirming test determination.

   Subscript *R* applied to the above symbol indicates value to be used in rating the unit.

7.62. *Tables of Physical Properties*

a. Heat and moisture content of air shall be determined from Tables 1 and 2 in Appendix 1.

b. Other physical data shall be obtained from International Critical Tables or the current issue of *A.S.R.E. Refrigerating Data Book.*

7.63. The primary test determination of the refrigerant heat rejection capacity of the evaporative condenser is as follows:

a. If a Refrigerant Calorimeter is used the rate of refrigerant flow is—

$$w_r = \frac{q_e}{h_{1e} - h_{le}} \text{ in lb. per hr.}$$

b. If a Refrigerant Liquid Quantity Meter is used the rate of refrigerant flow is—

$$w_r = \frac{Q_{rm}}{v_{rm}} \text{ in lb. per hr.}$$

c. The *Total Refrigerant Heat Rejection Effect* is—

$q_t = w_r\,(h_{sc} - h_{lc})$ in Btu. per hr.

d. The *Condensing Heat Rejection Effect* is—

$q_c = w_r\,(h_{sc} - h'_{lc})$ in Btu. per hr.

e. The *Subcooling Refrigerant Heat Rejection Effect* is—

$q_s = q_t - q_c$ in Btu. per hr.

7.64. The confirming test determination of the refrigerant heat rejection capacity of the evaporative condenser is as follows:

a. If the air flow measurement apparatus method is used:

   1. The volumetric air flow rate at the unit discharge opening is—

$$Q_{ad} = 1098\, A c_d \left[\frac{p_{av}\, v'_{ad}}{1 + C_{ad}} \right]^{0.5} \text{ in cu. ft per min.}$$

7.64.

where

$$v'_{ad} = \frac{0.754\,(t_{ad} + 459.6)}{p_{ab}} \left[1 + \frac{7000}{4360} C_{ad}\right]$$

2. The weight flow rate in pounds per minute of dry air is—

$$w'_{a} = \frac{Q_{ad}}{v'_{ad}}$$

3. The heat removed by the air stream is $q_{a} = 60w'_{a}\,[\Sigma_{ad} - \Sigma_{ae} + C_{ad}\,(t'_{ad} - t_{we}) - C_{ae}\,(t'_{ae} - t_{we})]$ in Btu. per hr.
4. The heat removed by the water discharged from the unit is— $q_{w} = w_{wd}\,(t_{wd} - t_{we})$ in Btu. per hr.
5. The heat added by the fan motor is — $q_{fm} = 3.42\,eP_{fm} - q_{k}$, in Btu. per hr.
   Where e is (1) the efficiency of the motor and drive if the motor and drive are not in the air stream and are outside of unit or (2) unity if the motor is in the air stream or inside the unit.
   q_{k} is the kinetic energy of the discharge air stream.
6. The kinetic energy of the discharge air stream

$$q_{k} = 0.4\,\frac{Q_{ad}\,(1 + C_{ad})}{v'_{ad}} \left(\frac{V}{1098}\right)^{2}$$ in Btu. per hr.

   Where V is the velocity of discharge air at the point of discharge from the evaporative condenser. The kinetic energy of the discharge air stream is very small for normal velocities, and may be neglected.
7. The heat added by the pump motor is $q_{pm} = 3.42\,eP_{pm}$ in Btu. per hr.
   Where e has the same meaning as in the preceding sub-paragraph.
8. The heat leakage losses in or out of the insulated unit are small, and may be neglected.
9. The confirming test refrigerant heat rejection capacity of the unit is $q_{te} = q_{a} + q_{w} - q_{fm} - q_{pm}$, in Btu per hr.

b. If the refrigerant liquid quantity or flow meter method is used the determination of the total refrigerant heat rejection effect shall be made as described in paragraph 7.63.

c. The deviation between the results of the primary and confirming tests

$$\Delta q_{t} = 100\,\frac{(q_{te} - q_{t})}{q_{t}}$$

shall not be greater than ± five (5.0) per cent.

7.65. When a refrigerant liquid quantity or flow meter is used for the confirming capacity test, the test air flow rate is determined from Duct and Pitot Tube apparatus readings as follows:

$$Q_{ad} = 1098\,A \left[\frac{p_{av}\,v'_{ad}}{1 + C_{ad}}\right]^{0.5}$$ in ft per min.

7.66. Determination of the air flow rate in cu. ft. per min. of Standard Air is as follows:

$$Q_{ad}\ \text{Std} = \frac{Q_{ad}\,(1 + C_{ad})}{.07488\ v'_{ad}}$$

7.67. A correction for deviation from A.S.R.E. Standard Rating Conditions of the test values of entering air wet bulb and the entering refrigerant vapor shall be made as follows:
The correction factor determined by

$$F = \frac{t'_{se}\ \text{std.} - t'_{ae}\ \text{std.}}{t'_{se} - t'_{ae}}$$

shall be applied to test values so that,

$q_{eR} = q_{e}F$

$q_{sR} = q_{s}F$

These corrected values shall be used in determining *A.S.R.E. Standard Ratings.*

7.68. *A.S.R.E. Standard Ratings* shall be determined from test results as follows:

a. The *Condensing Refrigeration Effect* is— $\dfrac{q_{eR}}{1.20}$, in Btu. per hr.

b. The *Subcooling Refrigeration Effect*, by definition, is— q_{sR}, in Btu. per hr.

c. The total water supply in gal. per hr. shall be the test value.

d. The water overflow in gal. per hr. shall be the test value.

e. The total fan motor power input in watts or kilowatts shall be the test value.

f. The total pump motor power input in watts or kilowatts shall be the test value.

g. Air flow through the unit in cu. ft. per min. of *Standard Air* shall be the value determined from test data.

h. The value of external resistance for the rated air flow in inches of water gage shall be the test value.

Section 8. DEFINITIONS

8.01. An *Evaporative Condenser Unit* is a factory-made encased assembly of elements by which heat is transferred from refrigerant vapor and liquid to air and water, so that the refrigerant vapor is condensed or the refrigerant vapor condensed and the liquid subcooled. The air flow through the unit is mechanically produced, and the water quantity consumed is limited so that the cooling effect is essentially evaporative.

8.02. The *Refrigerant Compressor* is a mechanical device used in a refrigerating system for increasing the pressure of the refrigerant.

8.03. The *Total Refrigerant Heat Rejection Effect* is the total useful capacity of an evaporative condenser for removing heat from the refrigerant circulated through it. This is the product of the weight rate of refrigerant flow and the difference between the *Enthalpy* of the refrigerant vapor entering the unit and the *Enthalpy* of the refrigerant liquid leaving the unit, expressed in heat units per unit of time.

8.04. The *Condensing Heat Rejection Effect* is that portion of *Total Refrigerant Heat Rejection Effect* of an evaporative condenser that is used to condense the entering refrigerant vapor to a saturated liquid at the entering refrigerant pressure. This is the product of the weight rate of refrigerant flow and the difference between the *enthalpy* of the refrigerant vapor entering the unit and

8.04.

the *enthalpy* of a saturated liquid at the pressure of the refrigerant vapor entering the unit, expressed in heat units per unit of time.

8.05. The *Subcooling Refrigerant Heat Rejection Effect* is the *Total Refrigerant Heat Rejection Effect* less the *Condensing Heat Rejection Effect.* This is the product of the weight rate of refrigerant flow and the difference between the *enthalpy* of a saturated refrigerant liquid at the pressure of the refrigerant vapor entering the unit and the *enthalpy* of the refrigerant liquid at the leaving temperature, expressed in heat units per unit of time

8.06. The *Condensing Refrigeration Effect* is the *Condensing Heat Rejection Effect* less the heat added to the refrigerant vapor in the *Refrigerant Compressor.* This is the product of weight rate of refrigerant flow and difference between the *enthalpy* of the refrigerant vapor entering the *Refrigerant Compressor* and the *enthalpy* of the saturated refrigerant liquid at the pressure of the refrigerant vapor entering the evaporative condenser, expressed in heat units per unit of time. This value is a practical expression of the capacity of a refrigeration system of which the evaporative condenser is a component part.

8.07. The *Subcooling Refrigeration Effect* is the additional refrigeration effect made available by subcoolin the refrigerant liquid in the evaporative condenser. It value is identical to that of the *Subcooling Refrigeran Heat Rejection Effect.*

8.08. *Standard Barometric Pressure* is a barometric pressure of 29.92 in. of mercury.

8.09. *Standard Air* is air at a temperature of 68° F dry bulb and at *Standard Barometric Pressure* of 29.9 in. of mercury, which contains 0.00725 lb. of moisture pe lb. of dry air. *Standard Air* weighs 0.07488 lb. per cu ft.

8.10. *Enthalpy* is the sum of internal energy and pressure-volume product of a fluid, both expressed in hea units per unit weight. This quantity is sometimes calle "total heat" or "heat content."

8.11. *Evaporative Equilibrium* of a wet bulb instrumen is the condition attained when the wetted wick has reached a stable and constant temperature. When the instrument is exposed to air at velocities over 400 ft. pe min., this temperature may be considered the temperature of adiabatic saturation.

8.12. *Sigma Function* is the enthalpy of the air less that portion of the enthalpy required to raise the water content of the air in the liquid phase from the reference point to the temperature of adiabatic saturation.

Appendix I

le 1. Properties of Moist Air at Standard Barometer (29.92 in. Hg.) for Various Adiabatic-Saturation (Wet-Bulb) Temperatures t' and Dry-Bulb Temperatures t

p. of batic ation t'	*Vapor pressure, in. Hg.* e'	*Grains per lb. dry air* $(t-t')-0$ W'	*Sigma function, Btu.*	*(W′ — W) grains moisture per lb. dry air corresponding to (t — t′)*					
				10F.	*20F.*	*30F.*	*40F.*	*50F.*	*60F.*
50	0.3626	53.39	20.13	15.9	31.7	47.4	..	..	..
51	0.3674	55.4	20.67	15.9	31.7	47.4	..	..	..
52	0.3906	57.6	21.23	16.0	31.8	47.5	..	..	..
53	0.4052	59.8	21.81	16.0	31.8	47.5	..	..	.
54	0.4203	62.0	22.38	16.0	31.8	47.6	..	..	..
55	0.4359	64.35	22.97	16.0	31.9	47.6	63.3	..	..
56	0.4520	66.8	23.59	16.0	31.9	47.7	63.3	..	..
57	0.4686	69.3	24.20	16.0	32.0	47.8	63.4	..	..
58	0.4858	71.9	24.82	16.1	32.0	47.8	63.5	..	..
59	0.5035	74.5	25.46	16.1	32.0	47.9	63.6	..	..
60	0.5218	77.26	26.10	16.1	32.1	47.9	63.6	..	..
61	0.5407	80.1	26.75	16.1	32.1	48.0	63.7	79.3	..
62	0.5601	83.0	27.43	16.1	32.2	48.1	63.8	79.4	..
63	0.5802	86.1	28.12	16.2	32.2	48.1	63.9	79.5	..
64	0.6009	89.2	28.82	16.2	32.3	48.2	64.0	79.6	..
65	0.6222	92.47	29.56	16.2	32.3	48.3	64.1	79.8	..
66	0.6442	95.8	30.27	16.2	32.4	48.3	64.1	79.9	95.4
67	0.6689	99.3	31.06	16.3	32.4	48.4	64.2	80.0	95.6
68	0.6003	102.8	31.81	16.3	32.4	48.5	64.3	80.1	95.7
69	0.7144	106.5	32.61	16.3	32.5	48.5	64.4	80.2	95.9
70	0.7392	110.3	33.40	16.3	32.5	48.6	64.5	80.3	96.0
71	0.7648	114.3	34.24	16.4	32.6	48.7	64.6	80.4	96.1
72	0.7912	118.3	35.10	16.4	32.6	48.8	64.7	80.6	96.3
73	0.8183	122.4	35.94	16.4	32.7	48.8	64.8	80.7	96.4
74	0.8462	126.7	36.80	16.4	32.7	48.9	64.9	80.8	96.6
75	0.8750	131.2	37.71	16.5	32.8	49.0	65.0	81.0	96.7
76	0.9046	135.8	38.64	16.5	32.9	49.1	65.2	81.1	96.9
77	0.9352	140.5	39.57	16.5	32.9	49.2	65.3	81.2	97.1
78	0.9666	145.4	40.53	16.6	33.0	49.3	65.4	81.4	97.2
79	0.9989	150.4	41.50	16.6	33.0	49.3	65.5	81.5	97.4
80	1.0321	155.6	42.51	16.6	33.1	49.4	65.6	81.7	97.6
81	1.0664	160.9	43.53	16.7	33.2	49.5	65.8	81.8	97.8
82	1.1016	166.4	44.57	16.7	33.2	49.6	65.9	82.0	98.0
83	1.1378	172.1	45.67	16.7	33.3	49.7	66.0	82.2	98.2
84	1.1750	178.0	46.77	16.8	33.4	49.8	66.2	82.3	98.4
85	1.2133	184.1	47.91	16.8	33.4	49.9	66.3	82.5	98.6
86	1.2527	190.4	49.07	16.8	33.5	50.0	66.4	82.7	98.8
87	1.2931	196.9	50.26	16.9	33.6	50.1	66.6	82.9	99.0
88	1.3347	203.5	51.48	16.9	33.6	50.2	66.7	83.1	99.3
89	1.3775	210.3	52.72	17.0	33.7	50.4	66.9	83.2	99.5
90	1.4215	217.4	53.99	17.0	33.8	50.5	67.0	83.4	99.7
91	1.4667	224.7	55.30	17.0	33.9	50.7	67.2	83.6	99.9
92	1.5131	232.2	56.63	17.1	34.0	50.8	67.3	83.8	100.2
93	1.5608	239.9	57.99	17.1	34.0	50.9	67.5	84.1	100.4
94	1.6097	247.9	59.39	17.1	34.1	51.0	67.7	84.3	100.6
95	1.6600	256.0	60.84	17.2	34.2	51.2	67.9	84.5	100.9
96	1.7117	264.5	62.31	17.2	34.3	51.3	68.0	84.7	101.2
97	1.7647	273.2	63.82	17.3	34.4	51.4	68.2	84.9	101.5
98	1.8192	282.2	65.37	17.3	34.5	51.5	68.4	85.2	101.8
99	1.8751	291.5	66.96	17.4	34.6	51.6	68.6	85.4	102.1
100	1.9325	301.0	68.61	17.4	34.7	51.7	68.8	85.7	102.4

From "A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems" by W. H. Carrier and C. C. Mackey, in the A.S.M.E. ansactions for January, 1937.

Table 2. Additive Corrections to Standard Tabular Values of W′, W, and Σ for Deviations in Barometric Pressure from 29.92 in. Hg.

	ΔP_{ob} =	+1	ΔP_{ob} =	—1	ΔP_{ob} =	—2	ΔP_{ob}	—3	ΔP_{ob} =	—4
t′ °F.	ΔW′	ΔΣ	ΔW′	ΔΣ	ΔW′	ΔΣ	ΔW′	ΔΣ	ΔW′	ΔΣ
50	— 1.8	—0.27	1.9	0.28	3.9	0.59	6.0	0.92	8.3	1.27
51	— 1.8	—0.28	1.9	0.29	4.0	0.61	6.3	0.96	8.1	1.32
52	— 1.9	—0.29	2.0	0.30	4.2	0.64	6.5	0.99	9.0	1.37
53	— 2.0	—0.30	2.1	0.32	4.3	0.66	6.7	1.03	9.4	1.43
54	— 2.0	—0.31	2.2	0.33	4.5	0.68	7.0	1.07	9.7	1.48
55	— 2.1	—0.32	2.3	0.34	4.7	0.71	7.3	1.11	10.1	1.53
56	— 2.2	—0.33	2.3	0.35	4.8	0.73	7.6	1.15	10.5	1.59
57	— 2.3	—0.35	2.4	0.36	5.0	0.76	7.9	1.20	10.9	1.65
58	— 2.4	—0.36	2.5	0.38	5.2	0.79	8.2	1.24	11.3	1.71
59	— 2.5	—0.37	2.6	0.39	5.4	0.82	8.5	1.28	11.7	1.77
60	— 2.5	—0.38	2.7	0.41	5.6	0.85	8.8	1.33	12.2	1.84
61	— 2.6	—0.39	2.8	0.42	5.9	0.89	9.1	1.38	12.6	1.91
62	— 2.7	—0.41	2.9	0.44	6.1	0.92	9.5	1.43	13.1	1.98
63	— 2.8	—0.42	3.0	0.46	6.3	0.95	9.8	1.48	13.6	2.05
64	— 2.9	—0.44	3.2	0.48	6.5	0.98	10.2	1.53	14.1	2.13
65	— 3.1	—0.46	3.3	0.50	6.8	1.02	10.6	1.59	14.6	2.21
66	— 3.2	—0.48	3.4	0.51	7.0	1.06	11.0	1.65	15.2	2.29
67	— 3.3	—0.50	3.5	0.53	7.3	1.10	11.4	1.72	15.8	2.38
68	— 3.4	—0.51	3.7	0.55	7.5	1.14	11.8	1.78	16.3	2.46
69	— 3.5	—0.53	3.8	0.57	7.8	1.18	12.3	1.85	16.9	2.55
70	— 3.7	—0.55	3.9	0.59	8.1	1.22	12.7	1.91	17.5	2.64
71	— 3.8	—0.57	4.1	0.61	8.4	1.26	13.1	1.98	18.2	2.74
72	— 3.9	—0.59	4.2	0.63	8.7	1.31	13.6	2.05	18.9	2.84
73	— 4.1	—0.61	4.4	0.66	9.1	1.36	14.1	2.12	19.6	2.95
74	— 4.2	—0.63	4.5	0.68	9.4	1.41	14.6	2.19	20.3	3.05
75	— 4.4	—0.66	4.7	0.71	9.7	1.46	15.1	2.27	21.0	3.15
76	— 4.5	—0.68	4.9	0.73	10.1	1.51	15.7	2.35	21.7	3.25
77	— 4.7	—0.70	5.0	0.76	10.4	1.56	16.2	2.43	22.4	3.36
78	— 4.9	—0.73	5.2	0.78	10.8	1.62	16.8	2.52	23.2	3.48
79	— 5.0	—0.75	5.4	0.81	11.2	1.67	17.4	2.61	24.1	3.61
80	— 5.2	—0.78	5.6	0.84	11.6	1.73	18.0	2.70	25.0	3.74
81	— 5.4	—0.81	5.8	0.87	12.0	1.79	18.7	2.79	25.9	3.88
82	— 5.6	—0.84	6.0	0.90	12.5	1.86	19.3	2.89	26.8	4.01
83	— 5.8	—0.87	6.2	0.93	12.9	1.92	20.0	2.99	27.8	4.15
84	— 6.0	—0.90	6.4	0.96	13.3	1.99	20.7	3.10	28.8	4.29
85	— 6.2	—0.93	6.7	1.00	13.8	2.06	21.5	3.21	29.8	4.44
86	— 6.4	—0.96	6.9	1.03	14.2	2.13	22.3	3.32	30.9	4.60
87	— 6.7	—1.00	7.1	1.06	14.7	2.20	23.0	3.43	32.0	4.76
88	— 6.9	—1.03	7.4	1.10	15.3	2.28	23.9	3.55	33.1	4.93
89	— 7.1	—1.06	7.6	1.13	15.8	2.36	24.7	3.68	34.3	5.10
90	— 7.4	—1.10	7.9	1.17	16.4	2.44	25.6	3.81	35.5	5.28
91	— 7.6	—1.13	8.2	1.21	11.0	2.53	26.5	3.94	36.8	5.47
92	— 7.9	—1.17	8.5	1.26	17.6	2.62	27.4	4.08	38.1	5.66
93	— 8.2	—1.22	8.8	1.31	18.2	2.71	28.4	4.22	39.4	5.86
94	— 8.5	—1.27	9.1	1.35	18.8	2.80	29.4	4.37	40.8	6.07
95	— 8.8	—1.31	9.4	1.40	19.5	2.90	30.4	4.52	42.2	6.28
96	— 9.1	—1.35	9.8	1.45	20.2	3.00	31.5	4.68	43.7	6.50
97	— 9.4	—1.40	10.1	1.50	21.0	3.11	32.6	4.85	45.3	6.73
98	— 9.7	—1.44	10.4	1.55	21.7	3.21	33.7	5.02	46.9	6.96
99	—10.1	—1.49	10.8	1.60	22.4	3.32	34.9	5.19	48.5	7.20
100	—10.4	—1.54	11.1	1.65	23.2	3.43	36.1	5.37	50.2	7.44

Note: ΔW is about 1 per cent less than ΔW′ for $t - t' = 23$F. and proportionally less for other values of $(t - t')$.

From "A Review of Existing Psychrometric Data in Relation to Practical Engineering Problems" by W. H. Carrier and C. C. Mackey, in the A.S.M.E. Transactions for January, 1937.

Example of The Use of Tables 1 and 2

required to determine the sigma function (Σ) and ire content in grains per lb. of dry air (W) of lly saturated air at a dry bulb temperature (t) of F., a wet bulb temperature (t') of 90° F., and a etric pressure (p_{ab}) of 28.92 in. of mercury.

le 1 gives the sigma function and moisture con- (W') of saturated air at the standard barometric ire of 29.92 in. of mercury. The sigma function id for any degree of saturation (wet bulb depres- but a correction must be made in the moisture it for partial saturation. Thus, in Table 1, the sigma on for air at 90° F. wet bulb (temperature of adia- saturation) is 53.99 Btu. per lb. The moisture it for saturated air is 217.4 grains per lb. Since it is d to determine the moisture content of air with . ($t-t'$) wet bulb depression, the deductive correc- $W-W'$, is found for $t-t' = 10°$ F. In this case the ction is 17.0 grains per lb. Thus the moisture content .4 — 17.0 = 200.4 grains per lb. for air at 100° F. dry and 90° F. wet bulb at the standard barometric pres-

ice it is desired to determine the sigma function moisture content of this air at a barometric pressure .92 in. of mercury ($\Delta p_{ab} = -1$), the further cor- ons indicated in Table 2 are necessary. The correc- in sigma function, $\Delta\Sigma$, for $\Delta p_{ab} = -1$ is + 1.17 per lb. Adding this value to the sigma function d in Table 1, the correct sigma function is 53.99 + = 55.16 Btu. per lb.

e correction in moisture content, $\Delta W'$, for deviations standard barometric pressure are indicated in Table saturated air only. The note at the bottom of the shows the difference between $\Delta W'$ and ΔW for rent amounts of wet bulb depression. For a wet bulb ession of 10° F., the error in assuming $\Delta W' = \Delta W$ be neglected in consideration of the number of sig- ant figures in the tabulated value of $\Delta W'$. On this , $\Delta w' = 7.9$, and the corrected moisture content e air is 200.4 + 7.9 = 208.3 grains* per lb. of dry

---

here are 7000 grains per lb. Divide grains of moisture (water) per lb. of dry air (W) by 7000 to obtain lb. of moisture per lb. air (C).

ppendix 2.—DERIVATION OF FORMULAS

$$Q_{ad} = 1098.Ac_d \left[\frac{p_{av}\ v'_{ad}}{1 + C_{ad}}\right]^{0.5}$$, cu. ft.

per min. of air and water vapor mixture.

The fundamental equation for the flow of incompressible fluids may be applied to air flow with negligible error when pressure changes are small:

$$V_s = [2gh]^{0.5} \qquad (1)$$

where

V_s = velocity of fluid in ft. per sec.

g = acceleration due to gravity, in ft. per second per sec.

h = fluid head, in ft.

It is necessary to restate this equation so that velocity is expressed in ft. per min. and fluid head is expressed as the product of pressure in in. of water and specific volume in cu. ft. of air and water vapor mixture per lb. of dry air.

thus:

$$V_s = \frac{V}{60}$$

$$h = \frac{62.43}{12}\, p\,v = 5.203\, \frac{p\ v'}{1 + C}$$

where

V = velocity of air and water vapor mixture in ft. per min.

60 = sec. per min.

p = pressure, in in. of water

v = specific volume of air and water vapor mixture, in cu. ft. of mixture per lb. of mixture

v' = "specific volume" of air and water vapor mixture, in cu. ft. of mixture per lb. of dry air

C = lb. of water vapor per lb. of dry air

$1 + C$ = lb. of air and water vapor mixture per lb. of dry air

62.43 = density of water at 39.2°F. (4.0°C.), in lb. per cu. ft.

12 = in. per ft.

By substituting these new units in equation (1) and assigning the value of 32.17 ft. per sec.^{2} to g, the following equation is obtained:

$$\frac{V}{60} = \left[\frac{2.33.17.5.203\ p:v'}{1 + C}\right]^{0.5}$$

$$V = 1098 \left[\frac{p\ v'}{1 + C}\right]^{0.5} \qquad (2)$$

Volumetric rate of fluid flow through an orifice is given by the formula:

$$Q = A\,c_d V \qquad (3)$$

where

Q = volumetric flow rate, in cu. ft. per min.

A = area of orifice, in sq. ft.

c_d = discharge coefficient of orifice

V = velocity of fluid, in ft. per min.

Substituting the expression for V given in formula 2 into formula 3:

$$Q = 1098\ Ac_d \left[\frac{p\ v'}{1 + C}\right]^{0.5}$$

2. $$v'_{ad} = \frac{0.754\ (t_{ad} + 459.6)}{p_{ab}} \left[\frac{1 + 7000\,C_{ad}}{4360}\right],$$

in cu. ft. of air and water vapor mixture per lb. of dry air. The general equation for a gas is

$$pv = RT \qquad (1)$$

where

p = absolute pressure in lb. per sq. ft.

v = specific volume in cu. ft. per lb.

R = gas constant.

T = absolute temperature = temperature in °F. plus 459.6.

The absolute pressure may be expressed in in. of mercury.

$$p = \frac{848.7}{12}\, p_{ab} = 70.73\ p_{ab}$$

where

848.7 = density of mercury, at 32° F. in lb. per ft.

12 = in. per ft.

p_{ab} = pressure in in. of mercury.

The gas constant for air is 53.34, and using this value, equation 1 may be rewritten so that

$$v = \frac{53.34\ (t + 459.6)}{70.73\ p_{ab}} = \frac{0.754\ (t + 459.6)}{p_{ab}} \qquad (2)$$

Equations (1) and (2) are for dry air. Normally air contains some water vapor and a formula to express the volume occupied by such air in cu. ft. of air and water vapor mixture per lb. of dry air is desired. This is given by the formula

$$v' = \frac{v(1 + C)}{s} \qquad (3)$$

where

v' = "specific volume" of air and water vapor mixture, in cu. ft. of mixture per lb. of dry air.

C = lbs. of water per lb. of dry air.

s = specific gravity of water vapor, expressed as to the ratio of the density of water vapor to the density of dry air at the same pressure and temperature.

The assumption that water vapor conforms to the gas laws may be used with negligible error in determining the specific gravity. On this basis the specific gravity of water vapor equals the ratio between the molecular weight of water vapor (18.016) and dry air (28.996). This gives a value of 0.6220. Assigning this value to s in equation (3), and substituting the expression for v found in equation (2).

$$v' = \frac{0.754\ (t + 459.6)}{p_{ab}} \left[\frac{1 + C}{0.6220} \right]$$

$$= \frac{0.754\ (t + 459.6)}{p_{ab}} \left[\frac{1 + 7000\ C}{4360} \right]$$

3. $q_a = 60w'_a\ [\Sigma_{ad} - \Sigma_{ae} + C_{ad}\ (t'_{ad} - t_{we}) - C_{ae}\ (t'_{ae} - t_{we})]$, in Btu. per hr.

The above formula represents the rate at which heat is added to an air stream when the moisture content of the air is being increased by evaporation of water, as in an evaporative condenser Ordinarily the rate at which heat is added to a fluid may be expressed as the product of the weight rate of fluid flow and the change in enthalpy of fluid during the process under consideration. Since air is defined as a mixture of various gases and water vapor, it will be seen that the rate at which heat is added to the air stream in an evaporative condenser cannot be found in this simple way. More air leaves the unit than enters, since water evaporated in the unit increases the amount of discharged air.

Also, while the enthalpies of the entering and leaving air ray be determined from physical measurements, the enthalpies thus determined are for fluids with different amounts of moisture content and hence one enthalpy cannot be properly subtracted from the other to determine the enthalpy change during the process.

To resolve these difficulties, requires: (a) the use of special terms for the expression of air flow rate and air enthalpy and (b) a means of correcting for
the changed moisture content of the air. The weig
rate of air flow is converted so as to be expresse
in terms of "dry air," through determination of th
moisture content. Since the term "dry air" exclud
the variable water vapor content, weight rate of "d
air" flow through an evaporative condenser can b
evaluated. To be consistent, enthalpy is also e
pressed in terms of dry air, but this does not elim
inate the fact that the entering and leaving e
thalpies are determined for different fluids, i.e. a
with differing amounts of water vapor. The pra
tical effect of this is that the enthalpy differen
between the leaving and entering air includes he
that has not been added to the air stream in its pa
sage through the unit. This results from the fa
that the enthalpy of air includes the enthalpy of i
moisture content from the reference point of 32°F
and hence the enthalpy of the leaving air include
the heat required to raise the water evaporated int
the air stream (during its passage through the unit
from 32°F. to the temperature at which it enters th
unit. This amount of heat obviously cannot be cre
ited to the air stream, and should be deducted. Thu
the correct formula for heat added to the air strea
in an evaporative condenser is

$$q_a = 60w'_a\ [h_{ad} - h_{ae} - (C_{ad} - C_{ae})\ (t_{we} - 32)]$$

where

q_a = heat added to air stream, in Btu. per hr.

60 = min. per hr.

w'_a = weight rate of air flow, in lb. of dry ai per min.

h_{ad} = enthalpy of leaving air, in Btu. per lb. o dry air.

h_{ae} = enthalpy of entering air, in Btu. per lb of dry air.

C_{ad} = moisture content of leaving air, in lb. o water per lb. of dry air.

C_{ae} = moisture content of entering air, in lb. o water per lb. of dry air.

t_{we} = temperature of water entering unit.

32 = reference point for water vapor enthalpy in °F.

Complete enthalpy tables for air would be ver
bulky, and so a more adaptable expression for hea
content of air is generally used in pyschrometry
This is the "sigma function" which may be define
as the enthalpy of air less the enthalpy of th
liquid of the contained water vapor. The practica
advantage of the sigma function is that it is constan
for a constant temperature of adiabatic saturatio
(constant wet bulb temperature), and thus permits
the evaluation of the heat content of air for al
degrees of saturation from a simple table or
psychometric chart. For this reason the sigma function is used in this Code.

Use of the sigma function does not obviate the
necessity of correcting for the different moisture
content of the air entering and leaving unit in the
formula for the heat added to the air stream in a
evaporative condenser. Since the sigma function includes the heat of vaporization and superheat but
not the heat of the liquid, the sigma function difference between the leaving and entering air fails t
account for all the heat added to the air stream
in the unit. The sigma function of the leaving air
does not include the heat required to raise the water
evaporated in the unit from the supply water tem-

erature to the leaving air wet bulb temperature, or does it include the heat required to raise the itial water content of the air from the entering et bulb temperature to the leaving wet bulb temerature. Providing a correction for this additional eat results in the following formula:

$$q_c = 60w'_a\ [\Sigma_{ad} - \Sigma_{ae} + (C_{ad} - C_{ae})\ (t'_{ad} - t_{we}) + C_{ae}\ (t'_{ad} - t'_{ae})]$$

$$= 60w'_a\ [\Sigma_{ad} - \Sigma_{ae} + C_{ad}\ (t'_{ad} - t_{we}) - C_{ae}\ (t_{ae} - t_{we})]$$

here

Σ_{ad} = sigma function of leaving air, in Btu. per lb. of dry air.
Σ_{ae} = sigma function of entering air, in Btu per lb. of dry air.
t'_{ad} = wet bulb temperature of leaving air, in °F.
t'_{ae} = wet bulb temperature of entering air, in °F.
q_c, C_{ad}, C_{ae}, and t_{we} have same meaning as in preceding formula.

$q_{fm} = 3.42\ e\ P_{fm} - q_k$, in Btu. per hr.

A wattmeter placed in the power leads of an electric motor driving a fan measures the total power input to the electric motor. The motor is not a perfect converter of electrical energy into mechanical energy, hence some of the electrical input is dissipated as heat in the motor; the drive between the motor and fan has friction losses, and further electrical energy is dissipated here as heat. Thus the net amount of mechanical power reaching the fan is the product of the electrical power input and the combined efficiency of the motor and drive.

Since it is desired to express the net power input to the fan in Btu. per hr., the constant 3.42 representing the number of Btu. per hr. per watt is necessary in this equation.

Of the mechanical energy imparted to the fan, it is necessary to evaluate that portion which increases the heat content of the air between the air temperature measuring instruments at the entering and discharge openings of the unit. This would consist of the mechanical energy dissipated as heat due to the inefficiency of the fan, and that used in compressing the air. The only portion of the mechanical energy delivered to the fan that does not increase the enthalpy of the air is that represented by the kinetic energy of the leaving air stream. Hence in the equation given above, the power represented by movement of the air stream, designated as q_k, is deducted from the product of the combined fan and drive efficiency and the electrical input to the fan motor. The formula thus formed gives the net amount of heat added to the air per unit of time due to operation of the fan.

$$q_k = 0.4\ Q_{ad}\ \frac{(1 + C_{ad})}{v'_{ad}} \left[\frac{V}{1098}\right]^3, \text{ in Btu. per hr.}$$

The general equation for the kinetic energy per unit time is

$$q_e = \frac{1}{2}\ \frac{w}{g}\ V^2_s \qquad (1)$$

where

q_e = kinetic energy rate (power) in foot pounds per sec.
w = rate of fluid flow, in lb. per sec.
g = acceleration of gravity, in ft. per sec. per sec.
V_s = velocity of fluid, in ft. per sec.

To use this formula the units must be changed, since it is desired to express the kinetic energy rate in Btu. per hr., the weight rate of flow in terms of cu. ft. per min. and specific volume, and the velocity in ft. per min. Thus:

$$q_e = \frac{778}{3600}\ q_k \qquad (2)$$

where

778 = foot-pounds per Btu.
3600 = sec. per hr.
q_k = Btu. per hr.

$$w = \frac{Q}{60v} = \frac{Q\ (1 + C)}{60v'} \qquad (3)$$

where

Q = cu. ft. per min. of air and water vapor mixture.
v = specific volume of air and water vapor mixture, in cu. ft. of mixture per lb. of mixture.
v' = "specific volume" of air and water vapor mixture, in cu. ft. of mixture per lb. of dry air.
C = lb. of water vapor per lb. of dry air.
60 = sec. per min.

$$V_s = \frac{V}{60}$$

where

V = velocity in ft. per min.
60 = sec. per min.

Substituting these new units and assigning the value of 32.17 feet per second per second to g:

$$\frac{778 \times q_k}{3600} = \frac{1}{2} \times \frac{Q\ (1 + C)\ V^2}{60v'} \times \frac{1}{32.17} \times \frac{V^2}{3600}$$

$$q_k = 0.00000033\ \frac{Q\ (1 + C)\ V^3}{v'}$$

$$= 0.4\ \frac{Q\ (1 + C)}{v'} \left[\frac{V}{1098}\right]^3$$

INDEX

	Pages and Sections
A.S.R.E. Standard Ratings	7.68
Air Flow Measuring Apparatus	4.40; 4.55; 6.40; 7.50p
Apparatus	Sec. 6
Calorimeters	4.25; 6.20
Calorimeter, Brine Tank	6.22
Calorimeter, Concentric Tube	6.23
Calorimeter, Secondary Refrigerant	6.24; 7.50m
Classification	2.10
Computation of Results	7.60 to 7.68
Conditions, Standard	3.21
Data to be Recorded	7.50
Definitions	Sec. 8
Fan Types	2.30
Formulas, Derivation of	Appendix 2, Page 17
Instruments	Sec. 5 and 7.20
Liquid Meters, Flow or Quantity	4.26; 4.50; 6.30; 7.50n; 7.50o
Manometers	6.68; 7.26; 7.46
Method of Rating	Sec. 3
Nozzles, Impact or Static	6.51; 6.54; 6.55; 6.64; 6.65; 7.45
Physical Properties	7.62
Pitot Tube	6.70
Properties of Moist Air	Table 1, Page 15 Table 2, Page 16 Examples, Page 17
Purposes	1.20
Ratings	3.10 to 3.22
Receiving Chambers	6.50; 6.60
Scope	1.10
Specifications	3.22
Symbols	7.61
Test Methods, Confirming	4.30; 4.40; 4.50; 7.64
Test Methods, Primary	4.20; 7.63
Test Methods, Standard	4.10
Test Procedure	Sec. 7
Water Usage	2.20

ASRE Standard No. 22
(Supersedes Nos. 22 and 22i)

ASRE STANDARD
METHODS OF RATING AND TESTING WATER-COOLED REFRIGERANT CONDENSERS

---

by

THE JOINT COMMITTEES

ON

RATING COMMERCIAL REFRIGERATING EQUIPMENT AND
RATING INDUSTRIAL REFRIGERATING EQUIPMENT

Sponsored by The American Society of Refrigerating Engineers

Approved by ASRE Council, June, 1948

This ASRE Standard replaces ASRE Standard $22, "Methods of Rating and Testing Water-Cooled Refrigerant Condensers," and Standard $22i, "Methods of Rating and Testing Water-Cooled Industrial Refrigerant Condensers."

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40th STREET NEW YORK 18, N. Y.

Price 40 Cents

Joint Committee on
Rating Commercial Refrigerating Equipment

GLENN MUFFLY, Springfield, Ohio, *General Chairman*
LOUIS S. MORSE, York, Pennsylvania, *Secretary*

For The American Society of Refrigerating Engineers (Sponsor)—
Glenn Muffly, Chairman
R. H. Tull
Frank R. Zumbro*

For the Air Conditioning & Refrigerating Machinery Association—
Donald French, Chairman
Louis S. Morse

For the American Society of Heating & Ventilating Engineers—
F. H. Faust, Chairman

Acknowledgment is made of most valuable assistance by the following non-members of the Joint Committee:

Lars C. Hanson
J. C. Consley

*Deceased

Joint Committee on
Rating Industrial Refrigerating Equipment

FRED OPHULS, New York, N. Y., *Chairman*

For The American Society of Refrigerating Engineers (Sponsor)—
R. J. Panlener
J. P. McShane
Fred Ophuls

For Air Conditioning and Refrigerating Machinery Association—
J. C. Consley
R. G. Ewer
Lars Hanson

For the American Institute of Chemical Engineers—
H. C. Duus

For the American Institute of Refrigeration—
G. A. Horne
Van R. H. Greene (Alternate)

For the American Meat Institute—
Allen McKenzie

For the American Petroleum Institute—
D. G. Brandt

For the American Society of Heating and Ventilating Engineers—
Scott F. Nicoll

For the Heating, Piping & Air Conditioning Contractors National Association—
Arnold M. Goelz
R. W. Waterfill (Alternate)

For the National Association of Practical Refrigerating Engineers—
C. T. Baker

For the National Association of Refrigerated Warehouses—
James C. Irwin

For the Refrigeration Equipment Manufacturers Association—
Oscar H. Buschman
W. L. Morrison

A.S.R.E. Standard Methods of
Rating and Testing Water-Cooled Refrigerant Condensers

ction 1. PURPOSE, CLASSIFICATION, AND DEFINITION OF TERMS

0 *Purpose*

1.11 This standard prescribes methods of rating and testing water-cooled industrial refrigerant condensers and provides for both absolute and comparative evaluation of the performance and capacity under specific, standardized conditions.

1.12 To attain this objective, this standard:
- a. Classifies water-cooled refrigerant condensers as to type.
- b. Lists and defines the terms used in the rating of refrigerant condensers.
- c. Establishes conditions and methods of test which shall be used for obtaining proposed *A.S.R.E. Standard Ratings* of water-cooled refrigerant condensers.
- d. Lists the conditions which should be specified for *Application Ratings.*

0 *Classification*

1.21 Types of water-cooled refrigerant condensers, as covered by this standard, are:
- a. Closed shell-and-tube.
- b. Closed shell-and-coil.
- c. Open shell-and-tube.
- d. Double-tube.

30 *Definition of Terms*

1.31 A *Water-Cooled Refrigerant Condenser* is a factory-made assembly of elements by which the flows of refrigerant vapor and water are maintained in such a heat transfer relationship that the refrigerant vapor is condensed into a liquid and the water is heated.

1.32 The *Total Refrigerant Heat Rejection Effect* is the total useful capacity of a water-cooled refrigerant condenser for removing heat from the refrigerant circulated through it. This is the product of the weight rate of refrigerant flow and the difference of enthalpy of the entering and leaving refrigerant fluid.

1.33 The *Condensing Heat Rejection Effect* is that portion of the *Total Refrigerant Heat Rejection Effect* of a condenser which is used for condensing the entering refrigerant vapor to a saturated liquid at the entering refrigerant pressure. This is the product of the weight rate of refrigerant flow and the difference between the enthalpy of the entering refrigerant vapor and that of a saturated refrigerant liquid at the same pressure.

1.34 The *Subcooling Refrigerant Heat Rejection Effect* is the *Total Refrigerant Heat Rejection Effect* less the *Condensing Heat Rejection Effect.* This is the product of the weight rate of refrigerant flow and the difference between the enthalpy of a saturated refrigerant liquid at the pressure of the entering refrigerant vapor and that of the refrigerant liquid at the leaving temperature. Where this value is three (3) per cent or less of the *Total Refrigerant Heat Rejection Effect,* it may be neglected for rating purposes. In such a case the *Total Refrigerant Heat Rejection Effect* may be substituted for the *Condensing Heat Rejection Effect.*

1.35 The *Condensing Refrigeration Effect* is the *Condensing Heat Rejection Effect* less the increase in enthalpy of the refrigerant vapor between the evaporator and the refrigerant condenser. This value is a practical expression of the capacity of a refrigeration system of which the condenser is a component part.

1.36 The *Subcooling Refrigeration Effect* is the additional refrigeration effect made available by subcooling the refrigerant liquid in the refrigerant condenser. Its value is identical with that of the *Subcooling Refrigerant Heat Rejection Effect.*

Section 2. RATINGS

2.10 *Terms Used in the Rating of Water-Cooled Refrigerant Condensers*

2.11 Ratings of water-cooled refrigerant condensers shall be expressed in the following terms:
- a. The *Condensing Refrigeration Effect,* in Btu. per hr.
- b. The *Subcooling Refrigeration Effect,* in Btu. per hr.
- c. Initial temperature of the condensing water, in °F.
- d. Water flow rate, in gal. per min.
- e. The water pressure drop through the condenser, in lb. per sq. in.

2.20 *Conditions for A.S.R.E. Standard Ratings*

2.21 *A.S.R.E. Standard Ratings* shall be given under at least one of the groups listed in Table 1.

Table 1

oup o.	*Saturated temperature of entering refrigerant vapor °F.*	*Minimum actual temperature of entering refrigerant vapor, °F.*	*Temperature of entering water, °F.*	*Temperature of leaving water, °F.*	*Temperature of ambient air, °F.*
	105	125	75	98	90
I	105	125	85	98	90
II	100	200	75	93	90
V	100	200	85	93	90

atio of *Condensing Heat Rejection Effect* to *Condensing Refrigeration Effect* shall be arbitrarily assumed as 1.20 for the above rating conditions.

Four groups of proposed A.S.R.E. rating conditions are shown in recognition of the variation that exists in practice for various applications and various refrigerants. When ratings are shown for but one group, it is assumed that that group will be selected which is most representative of the conditions of use for which the condenser is specifically offered.

2.30 *Conditions for Application Ratings*

2.31 Each set of Application Ratings should be accompanied by specifications of the following conditions:

a. Saturated temperature of the entering refrigerant vapor, °F.

b. Actual temperature of the entering refrigerant vapor, °F.

c. Temperatures of the entering and leaving water, °F.

d. Temperature of the ambient air, °F.

e. The ratio of *Condensing Heat Rejection Effect* to the *Condensing Refrigeration Effect* for each condition of saturated and actual temperatures of the refrigerant vapor entering and leaving the refrigeration compressor.

f. A statement of the normal refrigerant charge.

g. A statement of the refrigerant used.

h. A statement of the allowance made for fouling of surfaces on both the refrigerant and water sides, and for non-condensable gas.

2.32 *Published Capacity Ratings* shall include one or more *A.S.R.E. Standard Ratings* identified as such by group numbers, each with its associated cooling water consumption.

Section 3. TEST METHODS

3.10 *Standard Test Methods*

3.11 Standard test methods described below shall be used to determine *A.S.R.E. Standard Ratings.*

3.12 Tests shall consist of a primary test and a simultaneous confirming test under the conditions specified in 2.21.

3.13 The results of the confirming test shall be within plus and minus three (±3) per cent of the primary test, but the primary test shall govern for rating purposes.

3.14 Refrigerant flow through the condenser shall be produced by one of the following means:

a. Refrigeration compressor and low-side evaporator or calorimeter.

b. Refrigerant boiler.

3.20 *Primary Test Methods*

3.21 The primary test for closed condensers shall consist of measurement of the heat rejected from the refrigerant by determination of:

a. Heat rejected to the water.

b. Heat rejected through the external surfaces of the condenser to the ambient air.

3.22 The heat rejected from the refrigerant to the water is the product of the water flow rate and the enthalpy difference of the entering and leaving water.

3.23 The water flow rate shall be determined by use of a water quantity meter or water flow meter.

3.24 The enthalpy difference between the entering and leaving water shall be determined from temperature measurements.

3.25 Heat rejected through the external surfaces to the ambient air is the product of the mean temperature difference between condenser shell and the ambient air and the calculated overall hea transfer coefficient.

3.26 The heat rejected through the external surface of the condenser shall be held to three (3) pe cent or less of the heat rejected to the water Insulation material shall be used, if necessary

3.27 The primary test method for open shell-and-tube condensers shall be as given under 3.33-b.

3.30 *Confirming Test Methods*

3.31 The confirming test shall consist of a secondary measurement of the heat rejected from the refrigerant by determination of:

a. The product of enthalpy difference between the refrigerant fluid entering and leaving the condenser and the rate of refrigerant flow, where a complete refrigeration system is used to perform the test.

b. The heat added to the refrigerant fluid by the refrigerant boiler when this apparatus is used in place of a complete refrigeration system.

3.32 The enthalpy difference of the entering and leaving refrigerant fluid shall be determined from its pressure and temperature.

3.33 Rate of refrigerant flow shall be determined by use of one of the following apparatus:

a. Refrigerant calorimeter.

b. Water or brine cooler.

c. Refrigerant liquid quantity or flow meter.

3.34 A refrigerant calorimeter is an apparatus that provides a means of accurately measuring the amount of heat required to evaporate and superheat the refrigerant fluid flowing through it. The rate of refrigerant flow is given by dividing the heat input, including leakage losses, by the enthalpy difference of the entering and leaving refrigerant fluid.

3.35 The water or brine cooler shall be insulated to reduce heat leakage losses to one (1) per cent or less of the heat absorbed by the water or brine. The rate of refrigerant flow is given by dividing the heat input, including leakage losses, by the enthalpy difference of the entering and leaving refrigerant fluid.

Section 4. INSTRUMENTS, APPARATUS, AND TABLES OF PHYSICAL PROPERTIES

4.10 *Instruments*

4.11 Selection of instruments shall be made from the types listed in Section 14 or its later equivalent of the latest approved edition of the *Standard Methods of Rating and Testing Mechanical Condensing Units* of The American Society of Refrigerating Engineers.

4.12 Accuracy of instruments selected shall be in accordance with that specified in the latest approved edition of the *Standard Methods of Rating and Testing Mechanical Condensing Units* of The American Society of Refrigerating Engineers, excepting only that all pressure-measuring instruments shall have an accuracy within the limits of plus or minus one (±1) per cent of the absolute pressure readings.

4.13 All instruments shall be applied and used in accordance with the latest power test codes of the American Society of Mechanical Engineers applicable to them.

4.20 *Apparatus*

4.21 Apparatus shall be selected from those listed and calibrated in the manner specified in the latest approved edition of the *Standard Methods of Rating and Testing Mechanical Condensing Units* of The American Society of Refrigerating Engineers.

0 *Table of Physical Properties*

4.31 Physical data shall be obtained from the current issue of the *Refrigerating Data Book* or from the *International Critical Tables.*

Section 5. TEST PROCEDURE

0 *Apparatus Assembly*

5.11 The refrigerant condenser and the selected instruments and apparatus are assembled, connected, tested, dehydrated, evacuated and charged with normal amount of refrigerant. Liquid subcoolers and vapor superheaters shall be used, if necessary, to assure accuracy of measurements and to obtain rating conditions.

20 *Operation and Limits*

5.21 Start the system and obtain and maintain the conditions specified in Table 1 for the group selected in accordance with the following tolerances and instructions:

a. The arithmetic average of all pressure readings shall not vary from the specified values by more than two (2) per cent of their absolute values, nor shall individual values vary by more than this amount from the average value.

b. The arithmetic average of all temperature readings, excepting the actual entering refrigerant vapor temperature and the ambient air temperature, shall not vary from the specified values by more than one (1) °F., nor shall the individual values vary more than this amount from the average value. The excepted temperature readings shall not vary from the specified values by more than three (3) °F.

c. Temperature measurements used for determination of refrigerant fluid enthalpies shall be taken under conditions of vapor superheat or liquid subcooling that will permit their accurate determination.

5.22 After establishment of steady flow conditions, all required readings shall be taken every twenty minutes and the test shall be continued until four consecutive sets of readings are within the specified limits.

5.23 The test report shall include the date, observers' names, essential identifying physical data on the condenser tested, refrigerant used, all test readings, computations, and the determined rating values.

.30 *Computation of Results*

5.31 From the primary test determination, the *Total Refrigerant Heat Rejection Effect*, q_t is equal to the heat rejected to the water plus the heat leakage losses through the condenser external surfaces, except in the case of the open shell-and-tube condenser where the calculation shall be as in 5.32.

5.32 From the confirming test determination, the *Total Refrigerant Heat Rejection Effect*, q_{to} is equal to the product of the weight rate of flow, w_r, and the difference in enthalpy between the entering and leaving refrigerant fluid, Δh.

5.33 The deviation between the results of the primary and confirming tests,

$$100 \frac{(q_{to} - q_t)}{q_t},$$

shall not be greater than plus or minus three (±3) per cent.

5.34 The *Condensing Heat Rejection Effect*, q_w, is found by determining the weight rate of refrigerant flow, w_r, from the primary test,

$$w_r = \frac{q_t}{\Delta h}$$

then $q_w = w_r \Delta h$

where Δh is the difference in the enthalpy of the entering refrigerant vapor and that of the saturated liquid at the same pressure.

5.35 The *Subcooling Refrigerant Heat Rejection Effect* is

$$q_s = q_t - q_w$$

5.36 A correction for deviation from *A.S.R.E. Standard Rating Conditions* of the test values of entering and leaving water temperatures and the saturated temperature of the entering refrigerant vapor shall be made as follows:

The correction factor f is determined by

$$f = \frac{t_l \text{ std} - t_e \text{ std}}{t_l \text{ test} - t_e \text{ test}} \times \frac{\log \left[\dfrac{t'_r \text{ test} - t_e \text{ test}}{t'_r \text{ test} - t_l \text{ test}} \right]}{\log \left[\dfrac{t'_r \text{ std} - t_e \text{ std}}{t'_r \text{ std} - t_l \text{ std}} \right]}$$

where

t'_r = saturated temperature of entering refrigerant vapor, °F.
t_e = temperature of entering water, °F.
t_l = temperature of leaving water, °F.

5.37 *A.S.R.E. Standard Ratings* shall be determined from test results as follows:

a. The *Condensing Heat Rejection Effect* in Btu. per hour is $q_{wr} = f\, q_w$

b. The *Subcooling Heat Refrigeration Effect* is. by definition: q_{sr} in Btu. per hr.

c. The water flow rate in gal. per min. shall be the test value.

d. The water pressure drop through the condenser, in lb. per sq. in., shall be the test value.

ASRE Standard No. 23-R

[redacted] *ASRE Standard*

METHODS OF RATING AND TESTING REFRIGERANT COMPRESSORS

APPROVED BY A.S.R.E.

DEC 6 1949

[redacted]

The American Society of Refrigerating Engineers

[redacted]

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40th STREET NEW YORK 18, N. Y.

[redacted]

Committee on Revision of Standard No. 23

Fred Ophuls, *Chairman*

J. C. Consley
R. G. Ewer
Lars Hanson
H. C. Duus
K. E. Wolcott
D. G. Brandt
S. F. Nicoll
J. P. McShane
C. C. Smith
A. M. Goelz
R. W. Waterfill
C. T. Baker
J. C. Irwin
O. H. Buschmann
C. E. Ploeger

ASRE Standards Committee, 1949

Leon Buehler, Jr., *Chairman*

R. E. Backstrom
F. Y. Carter
J. R. Chamberlain
O. C. Eckel
H. M. Hendrickson
W. W. Higham
A. B. Newton

Joint Committee on Rating Refrigerating Equipment

M. F. Tokach, *Chairman*

Fred Ophuls, *Vice Chairman*

ASRE Representatives:

M. W. Garland
W. W. Higham
S. R. Hirsch
J. P. McShane

Proposed ASRE Standard

Methods of Rating and Testing Refrigerant Compressors

SECTIONS

pose
sification
nition of Terms
ings
t Methods
ruments and Tables of Physical Properties
t Procedure
npressor Data to Be Recorded and Report of Test
ne Tank Calorimeter Method
centric Tube Calorimeter Method
11. Secondary Refrigerant Calorimeter Method
12. Flooded System Primary Refrigerant Calorimeter Method
13. Dry System Primary Ref'nt Calorimeter Method
14. Gaseous Refrigerant Flow Meter Method
15. Liquid Refrigerant Quantity Method
16. Liquid Refrigerant Quantity Meter and Flow Meter Methods
17. Condenser Method
18. Compressor Heat Balance Method

Section 1. PURPOSE

This standard prescribes methods of rating and ng refrigerant compressors and provides for both lute and comparative evaluation of their perance and capacity under specific, standardized itions.

To attain this objective, this standard:

Classifies refrigerant compressors as to type.

Lists and defines the terms used in the rating of frigerant compressors.

Establishes conditions and methods of test ich shall be used for obtaining proposed ASRE andard Ratings of refrigerant compressors.

Lists the conditions which should be specified r Application Ratings of refrigerant compressors der operating conditions varying from those degnated for ASRE Standard Ratings.

Section 2. CLASSIFICATION

Types of refrigerant compressors, as covered by standard, are:

Single stage reciprocating refrigerant comessors

Multistage reciprocating refrigerant compress

Centrifugal refrigerant compressors

Rotary refrigerant compressors

Section 3. DEFINITION OF TERMS

he following definitions apply to parts and terms in this standard:

Refrigerant Compressor is a specific machine for npressing refrigerant vapor.

Capacity of a compressor is the refrigerating ect produced by the change in total heat content ween the refrigerant liquid at a temperature responding to the pressure of the vapor leaving compressor and the refrigerant vapor entering compressor.

Saturated Refrigerant Suction Vapor Temperae entering the compressor shall be determined by the pressure as measured at a point not less than 8 pipe diameters of a straight run of pipe ahead of the compressor suction inlet and before the shut-off valve, if used.

3.14 *Saturated Refrigerant Discharge Vapor Temperature* leaving the compressor shall be determined by the pressure as measured at a point not less than 12 pipe diameters of a straight run of pipe on the compressor discharge outlet and after the shut-off valve, if used.

3.15 *Power Input Rating* is the power input under the conditions required in Paragraph 4.21, listed as follows:

- a. Compressor only: Bhp at the shaft of the power source
- b. Compressor and motor: Kw input to motor

3.16 *ASRE Standard Power Input Rating* is the power rating under the conditions specified in Paragraph 4.21 as follows:

- a. Compressor only: Bhp at the shaft of the power source
- b. Compressor and motor: Kw input to motor

3.17 *Performance Factor* of a refrigerant compressor is a ratio of its capacity to its power input as follows:

- a. Compressor only: Bhp per ton at shaft of the power source
- b. Compressor and motor: Btu per watt hr or Kw per ton

Section 4. RATINGS

4.10 *Terms Used in the Rating of Refrigerant Compressors*

4.11 Ratings of refrigerant compressors shall be expressed in the following terms:

- a. *Compressor Refrigeration Capacity* in Btu per hr or in tons
- b. *Saturated Refrigerant Suction Vapor Temperature* in deg F
- c. *Actual Suction Vapor Temperature* in deg F
- d. *Saturated Refrigerant Discharge Vapor Temperature* in deg F

e. *Power Input Rating* in bhp or kw
f. *Performance Factor* in bhp or kw per ton or Btu per watt hr

4.20 *Conditions for ASRE Standard Ratings*

4.21 ASRE Standard Ratings shall be given under the conditions specified in at least one of the groups listed in Table 1. In rating centrifugal compressors a head versus cfm curve shall be established, at rated speed, with suction superheats given in Table 1. Rating however shall be based upon saturated suction temperature conditions at the suction pressure corresponding to the saturated refrigerant vapor temperature of the group chosen.

Table 1.

Group no.	*Saturated ref vapor temp entering compressor, F*	*Ref vapor temp entering compressor, F*	*Saturated ref vapor temp leaving compressor, F*
I	—10	65	100
II	5	65	100
III	20	65	105
IV	40	65	110
V	—10	15	95
VI	5	30	95
VII	20	40	100
VIII	40	55	100
IX	—40	—15	95
X	—40	—15	40
XI	—40	—15	0
XII	—25	± 0	40
XIII	—25	± 0	5

Group IX is generally intended as a rating basis for a compound compressor and Groups X, XI, XII, and XIII for a booster compressor.

Thirteen groups of proposed ASRE rating conditions are shown in recognition of the variation that exists in practice for various applications and various refrigerants. When ratings are shown for but one group, it is assumed that that group will be selected which is most representative of the conditions of use for which the compressor is specifically offered.

4.30 *Conditions for Application Ratings*

4.31 Each set of Application Ratings should be accompanied by specifications of the following conditions:

a. Saturated temperature of the entering refrigerant vapor, deg F
b. Actual temperature of the entering refrigerant vapor, deg F
c. Saturated temperature of the leaving refrigerant vapor
d. A statement of the refrigerant used
e. A statement of the method of compressor cylinder cooling

4.32 Published Capacity Ratings shall include one or more ASRE Standard Rating identified as such by group number, each with its associated speed and power input rating

Section 5. TEST METHODS

5.10 *Standard Test Methods*

5.11 Standard test methods described below shall be used to determine ASRE Standard Ratings.

5.12 Tests shall consist of a primary test and a simultaneous confirming
fied in Paragraph 4.21.

5.13 Results of the confirming test shall be within
percent of the primary test except as noted in P
graph 18.13, but the primary test shall govern
rating purposes.

5.20 *Primary Test Methods*

5.21 Primary tests are as follows:

a. Brine tank calorimeter (Sec. 9)
b. Concentric tube calorimeter (Sec. 10)
c. Secondary refrigerant calorimeter (Sec. 11)
d. Flooded system primary refrigerant calorim
(Sec. 12)
e. Dry system primary refrigerant calorim
(Sec. 13)
f. Gaseous refrigerant flow meter (Sec. 14)

5.30 *Confirming Test Methods*

5.31 Confirming tests are as follows:

a. Liquid refrigerant quantity method (Sec. 15)
b. Liquid refrigerant quantity meter and fl
meter (Sec. 16)
c. Condenser method (Sec. 17)
d. Compressor heat balance method (Sec. 18)

5.40 *Values to Be Determined*

5.41 Values to be determined under the conditi
specified in Paragraph 4.21 are:

a. Refrigerating capacity in Btu per hr or in t
of 12,000 Btu per hr
b. Power input in hp at the shaft of the po
source

5.50 *Types of Capacity Measuring Apparatus*

5.51 A compressor calorimeter is an apparatus
determining the refrigerating capacity of a c
pressor by measuring the heat input required
balance the refrigerating effect produced in
evaporator by the compressor.

5.52 A brine tank calorimeter includes an evapora
and a heating unit submerged in an externally
sulated brine bath. The refrigerating effect p
duced in the evaporator by the compressor is b
anced by heat added to the brine bath.

5.53 A concentric tube calorimeter includes a h
exchanger for the transfer of heat from a fl
heating medium to a vaporizing refrigerant. T
refrigerating effect produced by the compressor
balanced by heat given up by the fluid heati
medium.

5.54 A secondary refrigerant calorimeter includes
evaporator located in the top and a heating u
located in the bottom of a pressure-tight contain
partially filled with a secondary refrigerant. T
refrigerating effect produced in the evaporator
the compressor is balanced by heat added to
secondary refrigerant.

5.55 A primary dry or flooded system refriger
calorimeter includes an evaporator with a heat
unit in direct thermal contact with a primary
frigerant. The refrigerating effect produced
the evaporator by the compressor is balanced
heat added to the primary refrigerant.

5.56 A gaseous refrigerant flow meter include
mechanism and supplementary devices for de
mining the instantaneous rate of circulation of
gaseous refrigerant. The refrigerating effect p
duced by the compressor is the product of
pounds of refrigerant gas circulated per hour
the difference between the heat content per po

refrigerant vapor entering the compressor and refrigerant liquid, at a temperature correspond- to the pressure of the refrigerant vapor leav- the compressor.

A liquid refrigerant quantity meter is an ap- atus for determining the refrigerating capacity a compressor by measuring flow of the refriger- . It includes hand operated measuring cylin- s or an integrating mechanism and supplemen- y devices for determining quantity of liquid rigerant circulated in a given period of time. e refrigerating effect produced by the compressor the product of the pounds of refrigerant circu- ed per hour and the difference between the heat ntent per pound of refrigerant vapor entering d of refrigerant liquid at a temperature corre- onding to the pressure of the refrigerant vapor ving the compressor.

A liquid refrigerant flow meter includes a mech- ism and supplementary devices for determining e instantaneous rate of circulation of liquid re- igerant. The refrigerating effect produced by e compressor is the product of the pounds of re- igerant liquid circulated per hour and the dif- rence between heat content per pound of refrig- ant vapor entering and of refrigerant liquid at temperature corresponding to the pressure of the frigerant vapor leaving the compressor.

A condenser calorimeter includes a condenser nd means for measuring the heat dissipated it.

The compressor heat balance makes use of the iom that the sum of the heat changes taking place ound a compressor is equal to zero. The heat uivalent of the power developed at the shaft of e compressor is balanced by the algebraic sum of e heat entering and leaving the compressor.

ction 6. INSTRUMENTS AND TABLES OF PHYSICAL PROPERTIES

General

Instruments, whose types and accuracies are sted below, shall be calibrated against secondary andards before and after each test.

All instruments shall be applied and used in ac- rdance with the latest power test codes of the merican Society of Mechanical Engineers appli- ble to them.

Physical data shall be obtained from the cur- nt issue of the *Refrigerating Data Book* or the *ternational Critical Tables.*

Temperature Measuring Instruments

Temperature measurements shall be made with e or more of the following instruments:
Mercury-in-glass thermometers
Thermocouples
Electric resistance thermometers
Accuracy shall be within the following limits:
Brine or water in calorimeters, ± 0.1 deg F
Water in condensers, when data are used in confirming tests, ± 0.1 deg F
Wet bulb temperatures, ± 0.1 deg F
All other temperatures, ± 0.5 deg F

ressure Measuring Instruments

Pressure measurements shall be made with one more of the following instruments:
Mercury column
b. Bourdon tube gage
c. Aneroid pressure gage
6.32 All pressure measuring instruments shall have an accuracy within the limits of ± 1 percent of the absolute pressure readings or differential pressure readings.

6.40 ***Electrical Measuring Instruments***

6.41 Electrical measurements shall be made with one or more of the following instruments:
  a. Indicating
  b. Integrating
6.42 Accuracy shall be within the following limits:
  a. Indicating instruments, 0.5 percent of full scale reading
  b. Integrating instruments, 1 percent

6.50 ***Refrigerant Flow Measuring Instruments***

6.51 Volume measurements shall be made with one or more of the following instruments:
  a. Liquid refrigerant quantity meter
  b. Liquid refrigerant flow meter
  c. Gaseous refrigerant flow meter
6.52 Accuracy shall be ± 1 percent.

6.60 ***Brine or Water Flow Measuring Instruments***

6.61 Volume measurements shall be made with one or more of the following instruments:
  a. Liquid quantity meter, measuring either weight or volume
  b. Liquid flow meter
6.62 Accuracy shall be ± 1 percent.

6.70 ***Speed Measuring Instruments***

6.71 Speed measurements shall be made with one or more of the following instruments:
  a. Revolution counter
  b. Tachometer
  c. Stroboscope
  d. Oscillograph
6.72 Accuracy shall be ± 0.25 percent.

6.80 Time measurements shall be made with apparatus whose accuracy is ± 0.1 percent.

6.90 Weight measurements shall be made with apparatus whose accuracy is ± 0.2 percent.

Section 7. TEST PROCEDURE

7.10 ***Preparation***

7.11 The compressor shall be placed in a room in which temperature can be controlled.
7.12 Provision shall be made to insure that no refrigerant vapor leaves the system.
7.13 Correct oil and refrigerant charges shall be placed in the proper locations in the refrigerating system and oil shall be maintained at normal operating level in the compressor.
7.14 Mechanical checks shall be made to insure refrigerating system gas tightness, correct belt tension, compressor valve tightness, bearing freeness, correct compressor rotation, and compressor mounting comparable to service conditions.

7.20 ***Operation and Limits***

7.21 Start the compressor and, if motor driven, regulate the voltage at the motor terminals to ± 2 percent of rated voltage.
7.22 Adjust low pressure side conditions to specified suction pressure at compressor ± 1 percent of absolute pressure.

7.23 Adjust the temperature of the suction vapor entering the compressor to the temperature specified in the group selected in Paragraph 4.21 ± 5 deg.

7.24 Adjust the pressure leaving the compressor to ± 1 percent of the absolute pressure corresponding to the saturated refrigerant vapor temperature specified in Paragraph 4.21.

Average ambient temperature shall be 90 F ± 2 deg F.

No abnormal air flow shall be permitted over the compressor.

7.25 Operate the compressor, before taking readings, until steady temperatures and mechanical equilibrium are established.

Section 8. COMPRESSOR DATA TO BE RECORDED AND REPORT OF TEST

8.10 Compressor data to be recorded shall include:
- 8.11 Pressure of refrigerant vapor entering the compressor stop valve
- 8.12 Temperature of refrigerant vapor entering the compressor stop valve
- 8.13 Pressure of refrigerant gas leaving the compressor stop valve
- 8.14 Temperature of refrigerant gas leaving the compressor stop valve
- 8.15 Ambient temperature of air at compressor
- 8.16 Revolutions per minute of compressor
- 8.17 Barometric pressure
- 8.18 For additional compressor data to be recorded for heat balance method see Paragraph 18.31.

8.20 Report of test shall include:
- 8.21 Test number
- 8.22 Date of test
- 8.23 Compressor manufacturer's name
- 8.24 Compressor model designation
- 8.25 Motor nameplate rating
- 8.26 Compressor test speed
- 8.27 Compressor displacement in cubic inches per revolution
- 8.28 Refrigerant name
- 8.29 ASRE rating group
- 8.31 Compressor capacity, Btu per hr or tons
- 8.32 Motor input, watts or kilowatts
- 8.33 Brake hp of power source from calibration curve
- 8.34 List of compressor accessories regularly furnished and in place during the test

Section 9. BRINE TANK CALORIMETER METHOD

9.10 *Description*

9.11 Evaporator shall be of the dry system coil type with sufficient surface to superheat the refrigerant vapor leaving the calorimeter to approximately 65 F with a brine temperature not exceeding 100 F and yet being compact so that a minimum amount of brine will be required.

9.12 The refrigerant control may be either the hand regulated or the automatic expansion valve type and shall be located close to the brine tank. The control and the refrigerant line to the brine tank shall be insulated to minimize loss of refrigeration.

9.13 Heaters shall be of the electric immersion type with ample margin in capacity to provide ready control of the brine temperature.

9.14 The brine bath shall have a low enough freezing point to prevent it congealing at a temperature corresponding to the operating suction pressure.

9.15 Means shall be provided for agitating the b so that the maximum variation in tempera throughout the bath will not exceed 5 F. The added to the brine by agitation shall be determi

9.16 The tank shall be insulated in such a manner heat leakage will not be greater than 5 percen the capacity of the compressor.

9.20 *Calibration*

9.21 Calibration of the brine tank shall be perfor by the heat absorption method. This shall be ried out as follows:

a. Four temperature measuring instruments be used to indicate the ambient temperat around the calorimeter, each instrument be located approximately in the middle of one of vertical outside wall areas of the calorimeter spaced ½ in. from such wall. During calibrat the average ambient temperature indicated the average of the four temperature readi shall be held constant within ± 2 deg F at any sired value not exceeding 90 F.

b. Agitators shall be operated and calorim heater input adjusted to maintain average b temperature at approximately 25 deg F ab ambient temperature.

c. After thermal equilibrium has been establis the heat shall be held constant, and ambient brine temperatures shall be recorded each h for not less than 10 hr.

d. Mean temperature difference between brine ambient shall be determined from the differe between average brine temperatures and aver ambient temperatures during the period of th mal equilibrium.

e. Heat leakage coefficient is given by:

$$q_l = \frac{Q_h + Q_i}{t_b - t_a}$$

q_l = heat leakage coefficient, Btu per hr per F difference
Q_h = calorimeter heat input, Btu per hr
Q_i = calorimeter agitator heat added to bri Btu per hr
t_a = average ambient temperature around calc meter, deg F
t_b = average brine temperature, deg F

f. Heat leakage is given by:

$$Q_l = q_l \ (t_a - t_b)$$

q_l = heat leakage coefficient, Btu per hr per F difference
Q_l = heat leakage into calorimeter, Btu per
t_a = average ambient temperature, deg F
t_b = average brine temperature, deg F

g. This method is recommended, but the calc meter may be calibrated for heat leakage by other equivalent method.

9.30 *Procedure*

9.31 a. Proper suction pressure at the compress corresponding to the specified saturated refrig ant temperature, is adjusted by means of refrigerant control; and the specified temperat of the refrigerant vapor entering the compres is adjusted by varying the heat input to brine bath.

b. Proper discharge pressure at the compress corresponding to the specified saturated refr erant temperature, is adjusted by varying emperature and flow of the condensing medium.

Calorimeter data to be recorded, after com- ssor rating specified conditions have become es- lished, are:

Pressure of refrigerant vapor at evaporator outlet

Temperature of refrigerant vapor at evaporator outlet

Pressure of liquid refrigerant entering refrigerant control

Temperature of liquid refrigerant entering refrigerant control

Temperature of brine

Ambient temperature at calorimeter

Calorimeter heat input

Calorimeter agitator heat input

All readings shall be taken every 30 min.

Test shall be continued until four successive adings are within the allowable limits given in aragraph 7.20 and variation in brine temperature etween the beginning and end of the test is not reat enough to cause an error of more than 1 per- nt in compressor capacity.

Calculations

Capacity of the compressor shall be determined rom the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}} (Q_h + Q_i + Q_l)$$

h_{f1} = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb

h_{f2} = heat content of refrigerant liquid entering refrigerant control, Btu per lb

h_{g1} = heat content of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, Btu per lb

h_{g2} = heat content of refrigerant vapor leaving calorimeter, Btu per lb

Q = compressor capacity, Btu per hr

Q_h = calorimeter heat input, Btu per hr

Q_l = calorimeter heat leakage gain, Btu per hr

Q_i = calorimeter agitator heat input, Btu per hr

on 10. CONCENTRIC TUBE CALORIMETER METHOD

) *Description*

11 A concentric tube calorimeter consists of two ndependent concentric fluid circuits, located in heat xchange relationship to each other. Liquid refrig- rant is introduced into the inner circuit where it s vaporized and the vapor superheated as pro- ided in Paragraph 4.21. A heating medium of nown properties is circulated through the outer ircuit where it supplies the heat required to vapor- ze and superheat the refrigerant.

12 The refrigerant control may be either the hand egulated or the automatic expansion valve type nd shall be located close to the calorimeter. The ontrol and the refrigerant line to the calorimeter hall be insulated to minimize loss of refrigeration.

13 The calorimeter shall be insulated in such a anner that heat leakage will not be greater than percent of the capacity of the compressor.

10.20 *Calibration*

10.21 Heat leakage of the calorimeter shall be determined by circulating the heating medium through the outer circuit of the calorimeter which is subject to no load other than heat leakage.

a. If water or brine is used, the temperature difference between water or brine into and out of the calorimeter shall be not less than 10 deg F and shall be established by regulating the rate of flow and the inlet temperature of the water or brine. Ambient temperature during this calibration shall be held constant within ± 2 deg F at any desired value not exceeding 90 F. Inlet temperature of the water or brine shall be at least 30 deg F above ambient temperature. The test shall be continued until four successive brine or water temperatures taken at one hour intervals at both water or brine inlet and outlet of the calorimeter and with constant rate of flow show fluctuations no greater than ± 0.2 deg F.

b. If steam is used as the heating medium, heat leakage of the calorimeter shall be determined by collecting and weighing the steam condensate from the heating medium circuit. The steam pressure shall be held constant within ± 0.5 lb per sq in. at any desired value. Ambient temperature during this calibration shall be held constant within ± 2 deg F at any desired value not exceeding 90 F. Precautions must be taken to maintain the steam surface fully active without air or water binding, to maintain not less than 8 deg F superheat on the steam admitted to the calorimeter, and to prevent loss by evaporation of condensate in the collecting vessel either by subcooling the condensate or adequately covering the vessel. Average outer surface temperature of the concentric tube shall be determined by at least ten equally spaced temperature measuring devices. The test shall be continued until four successive readings of condensate weight taken at not less than one hour intervals are within ± 10 percent of each other.

10.22 Heat leakage coefficient is given by:

a. If water or brine is used:

$$q_l = \frac{WC\ (t_1 - t_2)}{\frac{1}{2}\ (t_2 + t_1) - t_a}$$

C = specific heat of water or brine

q_l = heat leakage coefficient, Btu per hr per deg F difference

t_1 = temperature of water or brine entering calorimeter, deg F

t_2 = temperature of water or brine leaving calorimeter, deg F

t_a = average ambient temperature around calorimeter, deg F

W = weight of water or brine circulated, lb per hr

b. If steam is used:

$$q_l = \frac{(h_{s1} - h_{s2})\ W_s}{t_c - t_a}$$

h_{s1} = heat content of entering steam, Btu per lb

h_{s2} = heat content of steam condensate at leaving temperature, Btu per lb

q_l = heat leakage coefficient, Btu per hr per deg F difference
t_a = average ambient temperature around calorimeter, deg F
t_o = concentric tube average outer surface temperature, deg F
W_s = weight of steam condensate, lb per hr

10.23 Heat leakage is given by:

a. If water or brine is used:

$$Q_l = q_l\,[\tfrac{1}{2}\,(t_2 + t_1) - t_a]$$

Q_l = heat leakage from calorimeter, Btu per hr
q_l = heat leakage coefficient, Btu per hr per deg F difference
t_a = average ambient temperature around calorimeter, deg F
t_1 = temperature of water or brine entering calorimeter, deg F
t_2 = temperature of water or brine leaving calorimeter, deg F

b. If steam is used:

$$Q_l = q_l\,(t_c - t_a)$$

Q_l = heat leakage from calorimeter, Btu per hr
q_l = heat leakage coefficient, Btu per hr per deg F difference
t_a = average ambient temperature around calorimeter, deg F
t_c = concentric tube outer surface average temperature, deg F

10.24 This method is recommended, but the calorimeter may be calibrated for heat leakage by any other equivalent method.

10.30 *Procedure*

10.31 a. Proper suction pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by means of the refrigerant control; and the specified temperature of the refrigerant vapor entering the compressor is adjusted by varying the heat input to the calorimeter.

b. Proper discharge pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by varying the temperature and flow of the condensing medium.

10.32 If water or brine is used, temperatures of the water or brine entering and leaving the calorimeter during the test shall be maintained to within ±0.2 deg F. The weight of water or brine circulated through the calorimeter per hour shall be maintained within ±0.2 percent.

10.33 If steam is used as the heating medium, provision must be made to insure that only dry steam enters the calorimeter by maintaining not less than 8 deg F superheat. Steam pressure in the calorimeter shall be measured with a mercury column to an accuracy of ±2 percent, and temperature of entering steam and leaving condensate shall be read to an accuracy of ±0.5 deg F. Weight of condensate shall be measured to an accuracy of ± 1 percent, and precautions shall be taken to prevent loss of condensate by evaporation. Means must be provided to obtain an average surface temperature of the steam pipe.

10.34 Calorimeter data to be recorded, after compressor rating specified conditions have become established, are:

a. Pressure of refrigerant vapor at evaporator outlet
b. Temperature of refrigerant vapor at evapora
outlet
c. Pressure of liquid refrigerant before refrig
ant control
d. Temperature of liquid refrigerant before ref
erant control
e. Ambient temperature at calorimeter
f. Temperature of water or brine entering calori
ter, when water or brine is used
g. Temperature of water or brine leaving calori
ter, when water or brine is used
h. Weight of brine or water circulated per hr, w
water or brine is used
i. Temperature of steam entering calorimeter, w
steam is used
j. Pressure of steam in calorimeter, when steam
used
k. Temperature of condensate leaving calorime
when steam is used
l. Weight of condensate in lb per hr, when steam
used
m. Surface temperatures of steam pipe, when ste
is used

10.35 All readings shall be taken every 20 min.

10.36 Test shall be continued until four success
readings are within the allowable limits given
Paragraph 7.20 and variations in temperature a
pressure conditions of the heating medium dur
the test shall not be great enough to cause
error of more than 1 percent in compressor capaci

10.40 *Calculations*

10.41 Capacity of the compressor shall be determi
from the following:

a. When water or brine is used:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\,[WC\,(t_1 - t_2) - Q_l]$$

C = specific heat of water or brine
h_{f1} = heat content of refrigerant liquid at a te
perature corresponding to the pressure of
frigerant leaving compressor, Btu per lb
h_{f2} = heat content of refrigerant liquid enteri
refrigerant control, Btu per lb
h_{g1} = heat content of refrigerant vapor enteri
compressor, under conditions specified in Pa
graph 4.21, Btu per lb
h_{g2} = heat content of refrigerant vapor leavi
calorimeter, Btu per hr
Q = compressor capacity, Btu per hr
Q_l = calorimeter heat leakage loss to ambient
mosphere, Btu per hr
t_1 = temperature of water or brine entering c
orimeter, deg F
t_2 = temperature of water or brine leaving ca
rimeter, deg F
W = weight of water or brine circulated, lb p
hr

b. When steam is used:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\,[W_s\,(h_{s1} - h_{s2}) - Q_l]$$

h_{f1} = heat content of refrigerant liquid at a te
perature corresponding to the pressure of r
frigerant leaving compressor, Btu per lb
h_{f2} = heat content of refrigerant liquid enteri
refrigerant control, Btu per lb

$_{1}$ = heat content of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, Btu per lb

$_{g2}$ = heat content of refrigerant vapor leaving calorimeter, Btu per hr

$_{s1}$ = heat content of entering steam, Btu per lb

$_{s2}$ = heat content of steam condensate at leaving temperature, Btu per lb

$_l$ = calorimeter heat leakage loss to ambient atmosphere, Btu per hr

$_e$ = weight of steam condensate, lb per hr

tion 11. SECONDARY REFRIGERANT CALORIMETER METHOD

Description

The secondary refrigerant calorimeter shall nsist of a direct expansion coil or series of coils parallel serving as a primary evaporator. This aporator is suspended in the upper part of a essure-tight vessel. A secondary refrigerant is aced in the bottom of this vessel and arrange- ents are made to supply heat to the liquid.

2 The refrigerant control may be either the hand gulated or the automatic expansion valve type and ould be located close to the calorimeter. The con- ol and the refrigerant line to the calorimeter ould be insulated to minimize loss of refrigera- on.

3 The calorimeter shall be insulated in such a anner that heat leakage will not be greater than percent of the capacity of the compressor.

4 Means shall be provided to obtain the pressure the calorimeter secondary refrigerant. The gage sed for this purpose shall be such that it can be ccurately read to within 0.5 psi.

5 The calorimeter shall be equipped with a safety witch which will stop the flow of heat into the econdary refrigerant at a predetermined pressure.

Calibration

1 Heat leakage of the calorimeter shall be deter- ined by adjusting the heat input to the secondary efrigerant at a rate sufficient to maintain the econdary refrigerant at a pressure correspond- g to approximately 25 deg F above ambient. he ambient temperature shall be held con- ant within ± 2 deg F at any desired value not ceeding 90 F.

eat input to the secondary refrigerant shall be aintained constant, and pressure readings shall be ken at one hour intervals and continued until ur successive values of saturated temperature do ot differ by more than 1 deg F.

2 Heat leakage coefficient is given by:

$$q_l = \frac{Q_h}{t_s - t_a}$$

q_l = calorimeter heat leakage coefficient, Btu per hr per deg F difference

Q_h = calorimeter heat input, Btu per hr

t_a = ambient temperature, deg F

t_s = saturated secondary refrigerant temperature, deg F

3 Heat leakage is given by:

$$Q_l = q_l\ (t_a - t_s)$$

Q_l = heat leakage into calorimeter, Btu per hr

q_l = heat leakage coefficient, Btu per hr per deg F difference

t_a = average ambient temperature around calorimeter, deg F

t_s = saturated secondary refrigerant temperature, deg F

11.24 This method is recommended, but the calorimeter may be calibrated for heat leakage by any other equivalent method.

11.30 *Procedure*

11.31 a. Proper suction pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by means of the refrigerant control; and the specified temperature of the refrigerant vapor entering the compressor is adjusted by varying the heat input to the secondary refrigerant.

b. Proper discharge pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by varying the temperature and flow of the condensing medium.

11.32 Calorimeter data to be recorded, after compressor rating specified conditions have become established, are:

a. Pressure of refrigerant vapor at evaporator outlet
b. Temperature of refrigerant vapor at evaporator outlet
c. Pressure of liquid refrigerant entering refrigerant control
d. Temperature of liquid refrigerant entering refrigerant control
e. Ambient temperature at calorimeter
f. Pressure of secondary refrigerant
g. Heat input to secondary refrigerant

11.33 All readings shall be taken every 20 min.

11.34 Tests shall be continued until four successive readings are within the allowable limits given in Paragraph 7.20, and variation in heat input to the calorimeter during the test shall not be great enough to cause an error of more than 1 percent in compressor capacity.

11.40 *Calculations*

11.41 Capacity of the compressor shall be determined from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\ (Q_h + Q_l)$$

h_{f1} = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb

h_{f2} = heat content of refrigerant liquid entering refrigerant control, Btu per lb

h_{g1} = heat content of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, Btu per lb

h_{g2} = heat content of refrigerant vapor leaving calorimeter, Btu per lb

Q = compressor capacity, Btu per hr

Q_h = calorimeter heat input, Btu per hr

Q_l = heat leakage into calorimeter, Btu per hr

Section 12. FLOODED SYSTEM PRIMARY REFRIGERANT CALORIMETER METHOD

12.10 *Description*

12.11 The flooded system primary refrigerant calorimeter shall consist of a pressure-tight vessel or vessels in parallel, containing liquid refrigerant to which heat is supplied by heating elements in direct thermal contact with the liquid refrigerant.

12.12 The refrigerant control may be either the hand regulated or the automatic expansion valve type and should be located close to the calorimeter. The control and the refrigerant line to the calorimeter should be insulated to minimize loss of refrigeration.

12.13 The calorimeter shall be insulated in such a manner that heat leakage will not exceed 5 percent of the capacity of the compressor.

12.20 *Calibration*

12.21 Calibration of the calorimeter may be performed by either the heat absorption method or the small condensing unit method.

The heat absorption method shall be carried out as follows:

a. Calorimeter shall be filled with liquid refrigerant to its normal operating level and the liquid inlet and vapor outlet stop valves shall be closed. Holding the ambient temperature constant within ±2 deg F at any desired value not exceeding 90 deg F, supply enough heat to maintain a temperature approximately 25 deg F above ambient. Heat input to the refrigerant shall be maintained constant, and pressure readings shall be taken at one hour intervals and continued until four successive values of saturated temperature do not differ by more than 1 deg F.

b. Heat leakage coefficient is given by:

$$q_l = \frac{Q_h}{t_r - t_a}$$

q_l = calorimeter heat leakage coefficient, Btu per hr per deg F difference

Q_h = calorimeter heat input, Btu per hr

t_a = average ambient temperature around calorimeter, deg F

t_r = saturated refrigerant temperature, deg F

12.22 The small condensing unit method shall be carried out as follows:

a. With the calorimeter in a 90 deg F ambient and no load applied other than heat leakage, a small condensing unit is operated until steady conditions are reached. Then the condensed liquid refrigerant is collected and measured in volume measuring vessels by the method of Section 15 over a period of time sufficient to accumulate not less than a 6-in. height of liquid in the measuring vessel.

b. Heat leakage coefficient is given by:

$$q_l = \frac{Vd\ (h_{g2} - h_{f2})}{t_a - t_r}$$

d = density of liquid refrigerant, for temperature at which flow is measured, lb per cu ft

h_{f2} = heat content of refrigerant liquid entering refrigerant control, Btu per lb

h_{g2} = heat content of refrigerant vapor leaving calorimeter, Btu per lb

q_l = calorimeter heat leakage coefficient, Btu per hr per deg F difference

t_a = average ambient temperature around calorimeter, deg F

t_r = saturated refrigerant temperature, deg

V = flow of liquid refrigerant, cu ft per hr

12.23 Heat leakage is given by:

$$Q_l = q_l\ (t_a - t_r)$$

Q_l = heat leakage into calorimeter, Btu per

q_l = heat leakage coefficient, Btu per hr per F difference

t_a = average ambient temperature around orimeter, deg F

t_r = saturated refrigerant temperature, deg

12.24 These methods are recommended, but the orimeter may be calibrated for heat leakage by other equivalent method.

12.30 *Procedure*

12.31 a. Proper suction pressure at the compres corresponding to the specified saturated ref erant temperature, is adjusted by means of refrigerant control; and the specified temperat of the refrigerant vapor entering the compre is adjusted by varying the heat input to the orimeter.

b. Proper discharge pressure at the compres corresponding to the specified saturated refr erant temperature, is adjusted by varying temperature and flow of the condensing medi

12.32 Calorimeter data to be recorded, after co pressor rating specified conditions have bec established, are:

a. Pressure of refrigerant vapor at evapora outlet

b. Temperature of refrigerant vapor at eva rator outlet

c. Pressure of liquid refrigerant entering frigerant control

d. Temperature of liquid refrigerant enter refrigerant control

e. Ambient temperature at calorimeter

f. Calorimeter heat input

12.33 All readings shall be taken every 20 min.

12.34 Test shall be continued until four success readings are within the allowable limits given Paragraph 7.20 and variation in heat input to calorimeter during the test shall not be gr enough to cause an error of more than 1 perce in compressor capacity.

12.40 *Calculations*

12.41 Capacity of the compressor shall be determin from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\ (Q_h + Q_l)$$

h_{f1} = heat content of refrigerant liquid at a te perature corresponding to the pressure of frigerant vapor leaving compressor, Btu per

h_{f2} = heat content of refrigerant liquid enteri refrigerant control, Btu per lb

h_{g1} = heat content of refrigerant vapor enteri compressor, under conditions specified in Pa graph 4.21, Btu per lb

h_{g2} = heat content of refrigerant vapor leavi calorimeter, Btu per lb

Q = compressor capacity, Btu per hr

Q_h = calorimeter heat input, Btu per hr

Q_l = heat leakage into calorimeter, Btu per h

13. DRY SYSTEM PRIMARY REFRIGERANT CALORIMETER METHOD

)escription

The dry system primary refrigerant calorimeter imilar to the system described in Section 12 ex- that a dry type evaporator is used and the ·ssary modifications in procedure are made to ·lop the full accuracy of this method.

The dry system primary refrigerant calorimeter ll consist of a series of pressure-tight tubular sels of suitable length and diameter for accomhing evaporation of the refrigerant circulated by compressor on test. Tubular vessels shall be elecally isolated and equipped with electrical heat- means such as: tube-type electric heaters with- the tubular vessels, wire-type heaters wound ut the tubular vessels, heaters thermally well nded to the tubular vessels; or the heating may accomplished by passing current through the ular vessels themselves utilizing the heat thus erated to evaporate the refrigerant.

The refrigerant control may be either the hand ulated or the automatic expansion valve type and uld be located close to the calorimeter. The con- l and the refrigerant line to the calorimeter shall insulated to minimize loss of refrigeration.

The calorimeter shall be insulated in such a nner that the heat leakage does not exceed 5 pert of the capacity of the compressor.

Surfaces of the pressure vessels and of the eleccal insulation of the heaters, if they are external the pressure vessels, and of bonding material, en bonding material is used, shall be supplied th at least 10 equally spaced temperature measur- devices for determining the mean surface temrature.

Calibration

Calibration of the calorimeter shall be perrmed by supplying enough heat to it to raise the mperature of the evaporator surface about 25 deg above the ambient. When internally located heats are used, the evaporator shall be filled with reigerating machine oil to serve as a heat transfer edium. In this case the oil shall be thoroughly eaned out after the calibration. The ambient mperature shall be held constant within ± 2 deg at any desired value not exceeding 90 deg F. Heat put to the calorimeter shall be maintained conant, and temperature readings shall be taken at e hour intervals and continued until four sucssive averages do not differ by more than 1 deg F.

2 Heat leakage coefficient is given by:

$$q_l = \frac{Q_h}{t_c - t_a}$$

q_l = calorimeter heat leakage coefficient, Btu per hr per deg F difference
Q_h = calorimeter heat input, Btu per hr
t_a = average ambient temperature around calorimeter, deg F
t_c = average surface temperature of tubes, and heaters if externally located, deg F

3 Heat leakage is given by:

$$Q_l = q_l\ (t_a - t_c)$$

Q_l = heat leakage into calorimeter, Btu per hr
q_l = heat leakage coefficient, Btu per hr per deg F difference

t_a = average ambient temperature around calorimeter, deg F
t_c = average surface temperature of tubes, and heaters if externally located, deg F

13.30 *Procedure*

13.31 a. Proper suction pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by means of the refrigerant control; and the specified temperature of the refrigerant vapor entering the compressor is adjusted by varying the heat input to the calorimeter.

b. Proper discharge pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by varying the temperature and flow of the condensing medium.

13.32 Calorimeter data to be recorded, after compressor rating specified conditions have become established, are:

a. Pressure of refrigerant vapor at evaporator outlet
b. Temperature of refrigerant vapor at evaporator outlet
c. Pressure of liquid refrigerant entering refrigerant control
d. Temperature of liquid refrigerant entering refrigerant control
e. Ambient temperature at calorimeter
f. Calorimeter heat input
g. Mean surface temperature of tubes, and heaters if externally located

13.33 All readings shall be taken every 20 min.

13.34 Test shall be continued until four successive readings are within the allowable limits given in Paragraph 7.20, and variation in heat input to the calorimeter during the test shall not be great enough to cause an error of more than 1 percent in compressor capacity.

13.40 *Calculations*

13.41 Capacity of the compressor shall be determined from the following:

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}}\ (Q_h + Q_l)$$

h_{f1} = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb
h_{f2} = heat content of refrigerant liquid entering refrigerant control, Btu per lb
h_{g1} = heat content of refrigerant vapor entering compressor under conditions specified in Paragraph 4.21, Btu per lb
h_{g2} = heat content of refrigerant vapor leaving calorimeter, Btu per lb
Q = compressor capacity, Btu per hr
Q_h = calorimeter heat input, Btu per hr
Q_l = heat leakage into calorimeter, Btu per hr

Section 14. GASEOUS REFRIGERANT FLOW METER METHOD

14.10 *Description*

14.11 The gaseous refrigerant flow meter shall consist of a nozzle or orifice plate for measuring, with an accuracy of 1 percent, the volume of refrigerant vapor flowing through it.

14.12 The flow meter shall be installed in the suction main of a closed system consisting of the refrigerant compressor, means for reducing the refrigerant pressure from the discharge to the suction level, means for removing excessive vapor superheat and returning the conditioned gas to the suction of the compressor.

14.13 The refrigerant control may be either the hand regulated type or automatic type pressure reducing valve located in the closed system.

14.14 Means for removing heat of compression shall be provided by tapping refrigerant vapor from the high pressure side of the system, liquefying it in a suitable condenser, and injecting the liquid refrigerant into the superheated refrigerant in the low pressure side of the system.

14.15 Specified compressor discharge pressure may be controlled by a hand or automatically operated valve located on the inlet side of the condensing medium to the condenser.

14.16 The liquid injection control may be either the manual type or the automatic type valve.

14.17 Refrigerant flow may be measured by use of an orifice plate or nozzle constructed and installed in accordance with procedures as outlined in the latest edition of the ASME *Power Test Code,* Part 5, Chapter 4, Flow Measurements by Means of Standardized Nozzles and Orifice Plates.

14.18 Location of the orifice plate or nozzle may be in either of two places depending on preference of operator.

a. Location of orifice plate or nozzle in system between point of injection of liquid refrigerant and compressor suction. Means must be provided to insure a homogeneous mixture of superheated gas at the orifice or nozzle.

b. Location of orifice or nozzle in hot gas line between the pressure reducing valve and the point of injection of liquid refrigerant into hot gas line. This location insures dry gas at metering device, but necessitates measurement of the refrigerant flow through the condenser.

14.20 *Calibration*

14.21 No calibration is required when ASME *Power Test Code* is followed.

14.30 *Procedure*

14.31 a. Proper suction pressure at the compressor, corresponding to the specified saturated refrigerant temperature, is adjusted by the pressure reducing valve; and the specified temperature of the refrigerant vapor entering the compressor is adjusted by varying the amount of liquid injected into the hot gas line.

b. Proper discharge pressure at the compressor, corresponding to the specified saturated refrigerant vapor temperature, is adjusted by varying the flow or temperature of the condensing medium.

14.32 Calorimeter data to be recorded, after compressor rating conditions have been established, are:

a. Temperature of refrigerant vapor at upstream side of metering device

b. Pressure of refrigerant vapor at upstream side of metering device

c. Pressure differential between upstream side and downstream side of metering device.

d. In the case where the orifice or nozzle is located as in 14.18-b, the following additional readings are necessary: (1) Temperature of refrig
vapor entering condenser; (2) Pressure o
frigerant vapor entering condenser; (3)
perature of liquid refrigerant leaving conde
and (4) Measurement of heat rejected in the
denser.

14.33 All readings shall be taken every 20 min.

14.34 Test shall continue until four successive
ings are within allowable limits.

14.40 *Calculations*

14.41 Capacity of the compressor shall be determ
from the following:

$$Q = (h_{g1} - h_{f1}) \times M$$

Q = compressor capacity, Btu per hr
h_{g1} = heat content of refrigerant vapor ente
compressor, under test conditions specifie
Paragraph 4.21, Btu per lb
h_{f1} = heat content of refrigerant liquid at a
perature corresponding to the pressure o
frigerant vapor leaving compressor, Btu pe
M = amount of refrigerant measured by or
lb per hr

14.42 Refrigerant circulation as measured by
orifice or nozzle shall be calculated in accord
with the methods as outlined in the latest editio
ASME *Power Test Code,* Information on Ins
ments and Apparatus, Part 5, Chapter 4, H
Measurement by Means of Standardized No
and Orifice Plates.

14.43 In the case of location of the metering devic
described in 14.18-b, the measurement of ref
erant flow will be the sum of the refrigerant
through the orifice or nozzle and the refrige
flow through the condenser. Flow through the
denser shall be calculated as follows:

$$M_l = \frac{Q_l}{(h_{g3} - h_{f3})}$$

The capacity of the compressor will then be:

$$Q = (h_{g1} - h_{f1})\ (M + M_l)$$

Q = compressor capacity, Btu per hr
Q_l = heat rejected in condenser, Btu per hr
h_{g1} = heat content of refrigerant vapor enter
compressor, under test conditions specified
Paragraph 4.21, Btu per lb
h_{g3} = heat content of refrigerant vapor enter
condenser, Btu per lb
h_{f1} = heat content of refrigerant liquid at a t
perature corresponding to the pressure of
frigerant vapor leaving compressor, Btu pe
h_{f3} = heat content of liquid refrigerant leav
condenser, Btu per lb
M = amount of refrigerant measured by orif
lb per hr
M_l = refrigerant flow through condenser, lb
hr

Section 15. LIQUID REFRIGERANT QUANTITY METHOD

15.10 *Description*

15.11 Apparatus for measuring liquid refrigerant v
ume shall consist of two vertical pressure vess
(about 4 ft high) connected in parallel by ho
zontal headers at the top and bottom, and sh

shut-off valves between the headers and vessels he lines entering and leaving each vessel. The els and their connections with the condenser the evaporator shall be thermally insulated. ssure is equalized at all times by means of a ll vapor line connecting the tops of the two sels. Recommended sizes of vessels are given in ble 2.

Table 2. Recommended Sizes of Refrigerant Vessels

ter,	Length, ft	Range of use, tons	Refrigerant
	4	0 to 2	Freon (F-12)
	4	0 ” 2	Sulfur dioxide
	4	0 ” 2	Methyl chloride
	4	2 ” 7½	Ammonia
	4	2 ” 10	Freon (F-12)
	4	2 ” 10	Sulfur dioxide
	4	2 ” 10	Methyl chloride
	4	7½ ” 40	Ammonia
	4	10 ” 25	Freon (F-12)
	4	10 ” 25	Sulfur dioxide
	4	10 ” 25	Methyl chloride

2 The liquid refrigerant line leaving each vessel all be equipped with a temperature measuring strument well.

3 Each vessel shall be fitted with a graduated ge glass covering approximately the entire length.

4 A valve shall be located in the liquid refrigerant e from the receiver to the calibrating vessels to gulate flow of liquid and maintain the normal erating level in the receiver.

Calibration

1 Each measuring vessel shall be calibrated with liquid of known density and the gage glass scales arked in accordance with the calibration.

2 Calibrations are recommended for various sizes vessels, as follows:

-in. diameter, in hundredths (1/100) cu ft
-in. diameter, in tenths (1/10) cu ft
-in. diameter, in fourths (1/4) cu ft

Procedure

1 Measurement of flow of liquid refrigerant is ade by timing the interval for the collection of a lume of liquid for a period of not less than two inutes. The diameter of the vessel used should such that the height of liquid refrigerant accuulated is not less than 6 in.

his volume is collected from the condenser serving e compressor by closing the outlet valve on the llecting vessel and the inlet valve on the other ssel which then continues the supply of refriger- t to the refrigerant control. Care must be taken start timing after a steady flow is established d to finish timing before the level in the vessel pplying the refrigerant control falls to a point hich breaks the liquid seal at the outlet. Tim- g shall be recorded on the vessel in which e liquid level is rising. Total volume of uid refrigerant in the two vessels must be nstant within 1 percent at all times during e test. This total volume shall be determined by simultaneous reading of the liquid levels in the ssels.

Data to be recorded, after compressor rating ecified conditions have become established, are:

a. Time for collection of liquid refrigerant volume
b. Temperature of liquid refrigerant in collecting vessel
c. Volume of liquid refrigerant collected

15.33 All readings should be taken every 20 min.

15.34 Test shall be continued until four successive readings are within the allowable limits given in Paragraph 7.20, and the variation in rate of liquid refrigerant flow during the test shall not be great enough to cause an error of more than 1 percent in compressor capacity.

15.35 Proportion of oil present in the liquid refrigerant shall be determined when operation of the system being tested has reached the conditions specified in Paragraph 15.34. A sample is withdrawn into a carefully weighed and evacuated vessel. Vessel and sample are then weighed. The refrigerant is evaporated and the vessel and remaining oil weighed. By deducting from both gross weights the net weight of the empty vessel, net weights of the mixture and of the oil are obtained.

15.40 *Calculations*

15.41 Capacity of the compressor shall be determined from the following:

$$Q = Vd\,[(1 - 0.018\,wd)\ \ (h_{g1} - h_{f1}) - 0.44\,w\,(t_{f1} - t_{g1})\ ]$$

d = density of liquid refrigerant, for temperature at which flow is measured, lb per cu ft
h_{f1} = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb
h_{g1} = heat content of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, Btu per lb
Q = compressor capacity, Btu per hr
t_{f1} = temperature corresponding to pressure of refrigerant vapor leaving compressor, deg F
t_{g1} = temperature of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, deg F
V = flow of mixture of liquid refrigerant and oil, cu ft per hr
w = weight of oil per unit weight of mixture of liquid refrigerant and oil, lb per lb

Section 16. LIQUID REFRIGERANT QUANTITY METER AND FLOW METER METHODS

16.10 *Description*

16.11 Liquid refrigerant meter may be either a quantity meter which consists of a metering mechanism accurate to within 1 percent for recording refrigerant flow on a suitable counter graduated in gallons or cubic feet, or a flow meter which consists of a mechanism with suitable supplementary devices for determining the instantaneous rate of flow of the liquid refrigerant in cubic feet or gallons per hour.

16.12 The liquid refrigerant quantity meter shall be connected in the liquid line between the liquid receiver outlet and the refrigerant control.

16.13 In order that the meter may function properly under all conditions and that it may be protected from abuse and low refrigerant charge, additional apparatus shall be provided, preferably as an integral part of the quantity meter assembly, as follows:

a. Subcooling coil ahead of the meter sufficient to prevent vaporization of refrigerant through the meter

b. Sight glasses located immediately before the subcooling coil and just beyond the meter to determine that vapor bubbles are not mixed with the liquid refrigerant

c. Bypass valve and tube for bypassing the liquid around the meter. This valve shall be open except when data are being recorded.

d. Thermometer and well for measuring the liquid refrigerant temperature prior to its entering the subcooling coil

e. Thermometer and well for measuring the temperature at which the liquid refrigerant enters the meter

f. Pressure gage connected to the outlet side of the meter

16.14 The calorimeter and its supply of heat load should be such as to balance out quickly and require little adjustment.

16.15 A supply of cold water at reasonably constant pressure and temperature together with a regulating valve should be provided for the meter subcooling coil.

16.20 *Calibration*

16.21 The meter shall be calibrated periodically.

16.22 Testing fluid used in calibrating the meter shall have an absolute viscosity not less than one-half nor more than twice that of the liquid refrigerant whose quantity it normally measures.

16.23 The meter shall be calibrated at not less than three rates of flow covering the capacity range used.

16.30 *Procedure*

16.31 With the meter bypass valve open, start the system in operation. After compressor rating specified conditions have become established, prepare for the test as follows:

a. Close the meter bypass valve.

b. Keep the liquid refrigerant subcooled at least 5 deg F below its saturation temperature at the outlet from the flow meter.

16.32 After the above specified conditions have been obtained, record data on the meter and the subcooling coil as follows:

a. Meter reading

b. Saturation temperature of the refrigerant determined from the gage connected to the outlet side of the meter

c. Temperature of the liquid entering the subcooling coil

d. Temperature of the liquid entering the meter

16.33 All readings shall be taken every 10 min.

16.34 Tests shall be continued until seven successive readings are within the allowable limits given in Paragraph 7.20 and the variation in rate of liquid refrigerant flow during the test shall not be great enough to cause an error of more than 1 percent in compressor capacity.

16.35 Proportion of oil present in the liquid refrigerant shall be determined when operation of the system being tested has reached the conditions specified in Paragraph 16.34. A sample is withdrawn into a carefully weighed and evacuated vessel. Vessel and sample are then weighed. Refrigerant is evaporated and the vessel and remaining oil weighed. By deducting from both gross weights the net weight of the empty vessel, net weights of the mixture and of the oil are obtained.

16.40 *Calculations*

16.41 Compressor capacity shall be determined
the following:

$$Q = Vd\,[(1 - 0.018wd)\ (h_{g1} - h_{f1}) \\ - 0.44w\ (t_{f1} - t_{g1})]$$

d = density of liquid refrigerant, for tem
ture at which flow is measured, lb per cu
h_{f1} = heat content of refrigerant liquid at a
perature corresponding to the pressure c
frigerant vapor leaving compressor, Btu
h_{g1} = heat content of refrigerant vapor ent
compressor, under the conditions specifi
Paragraph 4.21, Btu per lb
Q = compressor capacity, Btu per hr
t_{f1} = temperature corresponding to pressu
refrigerant vapor leaving compressor, deg
t_{g1} = temperature of refrigerant vapor ent
compressor, under conditions specified in
graph 4.21, deg F
V = flow of mixture of liquid refrigerant an
cu ft per hr
w = weight of oil per unit weight of mixtu
liquid refrigerant and oil, lb per lb

Section 17. CONDENSER METHOD

17.10 *Description*

17.11 The condenser is that which forms a part o
equipment used with the compressor being te
The calorimeter completes the equipment ne
except for the provisions that must be made
taking the readings listed in Paragraph 17.31.

17.12 This method shall be used only when the
culated heat leakage from the condenser to the
ambient is less than 2 percent of the refrigera
effect of the compressor.

17.20 *Calibration*

None.

17.30 *Procedure*

17.31 During the primary test, record shall be m
of:

a. Pressure of vapor refrigerant entering conde

b. Temperature of vapor refrigerant entering
denser

c. Pressure of liquid refrigerant leaving conde

d. Temperature of liquid refrigerant leaving
denser

e. Temperature of water entering condenser

f. Temperature of water leaving condenser

g. Ambient temperature surrounding condense

h. Quantity of condenser cooling water

i. Average temperature of external condenser s
face exposed to ambient if insulated

17.32 All readings shall be taken concurrently and
the same time intervals specified in the primary t
and shall be continued until four successive readi
are within the allowable limits given in Paragr
7.20. Variation in condensing water temperat
rise and rate of flow during the test shall not
great enough to cause an error of more than 1 p
cent in compressor capacity.

17.40 *Calculations*

17.41 Capacity of the compressor shall be calcula
as follows:

$$Q = \frac{h_{g1} — h_{f1}}{h_{g3} — h_{f3}} [W (t_2 — t_1) + Q_n]$$

$_{f1}$ = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb

$_{f3}$ = heat content of refrigerant liquid leaving condenser, Btu per lb

$_{g1}$ = heat content of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, Btu per lb

$_{g3}$ = heat content of refrigerant vapor entering condenser, Btu per lb

) = compressor capacity, Btu per hr

$_n$ = heat loss from condenser to surrounding air, Btu per hr, $S\ U_a\ (t_c — t_a)$ approximately

= outside surface of condenser, sq ft

$_a$ = ambient temperature, deg F

$_c$ = external surface temperature of condenser, deg F

$_1$= condenser entering water temperature, deg F

$_2$ = condenser leaving water temperature, deg F

$_a$ = Condenser to air heat transfer coefficient, Btu per hr per sq ft per deg F. A value of U_a = 2 should be used unless the actual value is determined experimentally.

W = flow of condenser cooling water, lb per hr

tion 18. COMPRESSOR HEAT BALANCE METHOD

Description

1 Algebraic sum of the heat quantities inherent the operation of any compressor is equal to zero. realize this test method, the readings listed in aragraph 18.31 are essential.

2 This method shall be used only when the callated or tested heat leakage from the compressor the 90 F ambient does not exceed 5 percent of e heat equivalent of the compressor bhp.

3 Results of this test shall be within ± 5 pernt of the primary test.

4 Correct temperature and pressure readings of s entering and leaving the compressor are necesry for accurately determining the respective gas alities. (See Paragraph 6.13)

5 This method is particularly adapted for a conming test with test method in Section 14.

Calibration

e.

Procedure

1 During the primary test, record shall be made :

Pressure of refrigerant gas entering compressor

Temperature of refrigerant gas entering compressor

Pressure of refrigerant gas leaving compressor

d. Temperature of refrigerant gas leaving compressor

e. Average ambient temperature surrounding compressor

f. Jacket water: Quantity; water temperature entering; and water temperature leaving

g. Lubricating oil cooler: Quantity of cooling water; water temperature entering; and water temperature leaving

h. Power required to operate compressor

18.32 All readings shall be taken concurrently and at the same time intervals specified in the primary test, and shall be continued until four successive readings are within the allowable limits given in Paragraph 7.20. Variations in temperature, pressure, and power readings during the test shall not be great enough to cause an error of more than 1 percent in compressor capacity.

18.40 *Calculations*

18.41 Capacity of the compressor shall be calculated as follows:

$$Q = \frac{h_{g1} — h_{f1}}{h_{g1} — h_{g3}} [2546 \times \text{bhp} — (Q_n + Q_j + Q_o)]$$

h_{f1} = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb

h_{g1} = heat content of refrigerant vapor entering compressor, under conditions specified in Paragraph 4.21, Btu per lb

h_{g3} = heat content of refrigerant vapor leaving compressor, Btu per lb

bhp = brake horsepower at shaft of compressor

Q = compressor capacity, Btu per hr

Q_n = heat loss from compressor to surrounding air, Btu per hr, $S\ U_a\ (t_o — t_a)$

Q_j = heat rejected in jacket water, Btu per hr, $W_j\ (t_2 — t_1)$

Q_o = heat rejected in lubricating oil, Btu per hr, $W_o\ (t_4 — t_3)$

S = outside surface of compressor, sq ft

t_a = ambient temperature, deg F

t_o = temperature of external surface of compressor, deg F

t_1 = temperature of water entering jacket, deg F

t_2 = temperature of water leaving jacket, deg F

t_3 = temperature of water entering oil cooler, deg F

t_4 = temperature of water leaving oil cooler, deg F

U_a = compressor to air heat transfer coefficient, Btu per hr per sq ft per deg F. A value of U_a = 2 should be used unless the actual value is determined experimentally.

W_j = flow of cooling water through compressor jacket, lb per hr

W_o = flow of cooling water through lubricating oil cooler, lb per hr

ASRE Standard No. 24
(Supersedes Nos. 24 and 24i)

A S R E STANDARD
METHODS OF RATING AND TESTING WATER AND BRINE COOLERS

by

THE JOINT COMMITTEE

ON

RATING COMMERCIAL REFRIGERATING EQUIPMENT AND
RATING INDUSTRIAL REFRIGERATING EQUIPMENT

Sponsored by The American Society of Refrigerating Engineers

Approved by ASRE Council, June, 1948

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40th STREET NEW YORK 18, N. Y.

Price 40 Cents

Joint Committee on
Rating Commercial Refrigerating Equipment

GLENN MUFFLY, Springfield, Ohio, *General Chairman*
LOUIS S. MORSE, York, Pennsylvania, *Secretary*

For The American Society of Refrigerating Engineers (Sponsor)—
Glenn Muffly Chairman
R. H. Tull
Frank R. Zumbro*

For the Air Conditioning & Refrigerating Machinery Association—
Donald French, Chairman
Louis S. Morse

For the American Society of Heating & Ventilating Engineers—
F. H. Faust, Chairman

Acknowledgment is made of most valuable assistance by the following non-members of the Joint Committee:

Lars C. Hanson
J. C. Consley
*Deceased

Joint Committee on
Rating Industrial Refrigerating Equipment

FRED OPHULS, New York, N. Y., *Chairman*

For The American Society of Refrigerating Engineers (Sponsor)—
R. J. Panlener
J. P. McShane
Fred Ophuls

For the Air Conditioning and Refrigerating Machinery Association—
J. C. Consley
R. G. Ewer
Lars Hanson

For the American Institute of Chemical Engineers—
H. C. Duus

For the American Institute of Refrigeration—
G. A. Horne
Van R. H. Greene (Alternate)

For the American Meat Institute—
Allen McKenzie

For the American Petroleum Institute—
D. G. Brandt

For the American Society of Heating and Ventilating Engineers—
Scott F. Nicoll

For the Heating, Piping & Air Conditioning Contractors National Association—
Arnold M. Goelz
R. W. Waterfill (Alternate)

For the National Association of Practical Refrigerating Engineers—
C. T. Baker

For the National Association of Refrigerated Warehouses—
James C. Irwin

For the Refrigeration Equipment Manufacturers Association—
Oscar H. Buschmann
W. L. Morrison

A.S.R.E. Standard Methods of Rating and Testing Water and Brine Coolers

on 1. PURPOSE, CLASSIFICATION, AND DEFINITION OF TERMS

urpose

.11 This standard prescribes methods for rating and testing of water and brine coolers under specific, standardized operating conditions and permits both absolute and comparative evaluation of their capacity and performance and provides a measure of their performance characteristics for the widely varying conditions under which they are used.

1.12 To attain this objective, this standard:
- a. Classifies water and brine coolers as to type.
- b. Lists and defines the terms used in the rating of water and brine coolers.
- c. Establishes conditions and methods of testing which shall be used for obtaining *A.S.R.E. Standard Ratings* of water and brine coolers.
- d. Lists the conditions which should be specified for *Application Ratings* of water and brine coolers under operating conditions varying from those designated for *A.S.R.E. Standard Ratings.*

Classification

1.21 Types of water and brine coolers, as covered by this standard, are:
- a. Closed shell-and-tube.
- b. Closed shell-and-coil.
- c. Open shell-and-tube.
- d. Double-tube.
- e. Open tank and coil.

Definition of Terms

1.31 A *Water or Brine Cooler* is a factory-made assembly of elements in which the water or brine and the refrigerant are in heat transfer relationship, causing the refrigerant liquid to evaporate and the water or brine to be cooled.

1.32 The *Total Refrigeration Effect* of the water or brine cooler is the product of the weight rate of refrigerant flow and the difference in enthalpy of the entering and leaving refrigerant fluid, expressed in heat units per unit of time.

1.33 The *Net Refrigeration Effect* is that portion of *Total Refrigeration Effect* of a water or brine cooler that produces useful cooling. This is the product of the weight rate of water or brine flow and the difference in enthalpy of the entering and leaving water or brine expressed in heat units per unit of time. It is represented, also, by the *Total Refrigeration Effect* less the heat leakage losses.

Section 2. RATINGS

2.10 *Terms Used in the Rating of Water or Brine Coolers*

2.11 Ratings of water or brine coolers shall be expressed in the following terms:
- a. The *Net Refrigeration Effect* in Btu. per hr. or in tons.
- b. Temperature of leaving water or brine.
- c. Saturated temperature of leaving refrigerant vapor.
- d. The water or brine flow rate in gal. per min.
- e. The water or brine pressure drop through the cooler, in lb. per sq. in.

2.20 *Conditions for A.S.R.E. Standard Ratings*

2.21 *A.S.R.E. Standard Ratings* shall be given under the conditions specified in at least one of the groups listed in Table I.

2.30 *Conditions for Application Ratings*

2.31 *Application Ratings* give capacity and performance under operating conditions other than those designated for *A.S.R.E. Standard Ratings.* Each set of *Application Ratings* shall be accompanied in some manner by specifications of the following conditions:
- a. Temperature of entering water or brine, °F.
- b. Temperature of leaving water or brine, °F.
- c. Saturated temperature of refrigerant vapor leaving cooler, °F.
- d. Actual temperature of leaving refrigerant vapor, °F.
- e. A statement of the normal refrigerant charge.
- f. A statement of the refrigerant used.
- g. Kind of brine and specific gravity at 60° F.

2.32 Published *Capacity Ratings* shall include one or more *A.S.R.E. Standard Ratings* identified as such by group numbers each with its associated brine or water flow rate.

Table 1.

Group No.	*Temperature of entering water or brine, °F.*	*Temperature of leaving water or brine, °F.*	*Saturated Temperature of leaving refrigerant vapor, °F.*	*Ambient air temperature, °F.*
I	59	44	40	90
II	56	46	40	90
III	52	47	40	90
IV	46	43	40	90
V	27	12	5	90
VI	24	14	5	90
VII	21	16	5	90
VIII	14	11	5	90

Section 3. TEST METHODS

3.10 *Standard Test Methods*

3.11 Standard test methods described below shall be used to determine *A.S.R.E. Standard Ratings.*

3.12 Tests shall consist of a primary test and a simultaneous confirming test under the conditions specified in Par. 2.21.

3.13 The results of the confirming test shall be within plus or minus three (±3) per cent of the primary test, but the primary test shall govern for rating purposes.

3.20 *Primary Test Methods*

3.21 The primary test shall consist of a measurement of the refrigeration effect produced in the cooler by determination of:

a. Heat removed from the water or brine.

b. Heat removed from the ambient air through the external surfaces of the cooler.

3.22 The *Net Refrigeration Effect* shall be determined by multiplying the weight rate of water or brine flow by the enthalpy difference of the entering and leaving water or brine.

3.23 The water or brine flow rate shall be determined by use of liquid quantity or flow meter.

3.24 The enthalpy difference between the entering and leaving water or brine shall be determined from temperature measurements.

3.25 Heat removed from the ambient air through the external surfaces of the cooler is the product of the mean temperature difference between the cooler and the ambient air and the calculated over-all heat transfer coefficient.

3.26 The heat removed from the ambient air shall be held, by use of insulation material if necessary, to one (1) per cent or less of the heat removed from the water or brine.

3.30 *Confirming Test Methods*

3.31 The confirming test shall consist of a secondary measurement of the refrigeration effect produced in the cooler by determination of the product of the weight rate of refrigerant flow and the enthalpy difference of the entering and leaving refrigerant fluid. This is the *Total Refrigeration Effect.*

3.32 Rate of refrigerant flow shall be determined by use of one of the following:

a. Closed water-cooled refrigerant condenser.

b. Refrigerant liquid quantity or flow meter.

3.33 The closed water-cooled refrigerant condenser shall be insulated, if necessary, so that the heat-leakage losses do not exceed three (3) per cent of heat rejected to the water. The rate of refrigerant flow is given by dividing the heat rejection, including leakage losses, by the enthalpy difference of the entering and leaving refrigerant fluid.

Section 4. INSTRUMENTS, APPARATUS, AND TABLES OF PHYSICAL PROPERTIES

4.10 *Instruments*

4.11 Instruments shall be selected from the types listed in the latest approved edition of the *Standard Methods of Rating and Testing Mechanical Condensing Units* of The American Society of Refrigerating Engineers.

4.12 Accuracy of instruments selected shall b
accordance with that specified in the latest
proved edition of the *Standard Methods of*
ing and Testing Mechanical Condensing U
of The American Society of Refrigerating
gineers, excepting only that all pressure-m
uring instruments shall have an accuracy
in the limits of plus or minus one (±1)
cent.

4.13 All instruments shall be applied and use
accordance with the latest power test code
the American Society of Mechanical Engin
applicable to them.

4.20 *Apparatus*

4.21 Apparatus shall be selected from those li
and calibrated in the manner specified in
latest approved edition of the *Standard M*
ods of Rating and Testing Mechanical Cond
ing Units of The American Society of Refri
ating Engineers.

4.30 *Tables of Physical Properties*

4.31 Physical data shall be obtained from the cur
issue of the *Refrigerating Data Book* or
International Critical Tables.

Section 5. TEST PROCEDURE

5.10 *Apparatus Assembly*

5.11 The water or brine cooler and the selecte
struments, apparatus, and compressor are
sembled, connected, tested, dehydrated, ev
ated, and charged with the normal amou
refrigerant. Liquid subcoolers, vapor su
heaters, or heat exchangers shall be use
necessary, to assure accuracy of measure
and to obtain rating conditions.

5.20 *Operation and Limits*

5.21 Start the system and obtain and maintai
conditions specified in Table 1 for the g
selected in accordance with the following
ances and instructions:

a. The arithmetic average of all pressure
ings shall not vary from the specified v
by more than two (2) per cent of their
lute values, nor shall the individual var
more than this amount from the ave
value.

b. The arithmetic average of all temper
readings, excepting the actual temper
of the refrigerant vapor leaving the c
and the ambient air temperature, shall
vary from the specified values by more
one-half (0.5) °F., nor shall the indiv
readings vary by more than this amount
the average value. The excepted temper
readings shall not vary from the spe
values by more than three (3) °F.

c. Temperature measurements used for
mination of refrigerant fluid enthalpies
be taken under conditions of liquid sub
ing or vapor superheat that will permit
accurate determination.

5.22 After establishment of steady flow condi
all required readings shall be taken every tw
minutes, and the test shall be continued
four consecutive sets of readings are withi
specified limits.

5.23 The test report shall include the date, ob
ers' names, essential identifying physical

of the cooler tested, refrigerant and brine used, all test readings and computations, and the determined rating values.

Computation of Results

5.31 From the primary test determination, the *Total Refrigeration Effect*, q_t, is equal to the *Net Refrigeration Effect* q_w, plus the heat removed from the ambient air, q_l. Thus,

$$q_t = q_w + q_l$$

5.32 From the confirming test determination, the *Total Refrigeration Effect*, q_{tc}, is equal to the product of the weight rate of refrigerant flow, w_{rc}, and the difference in enthalpy of the entering and leaving refrigerant fluid, Δh.

5.33 The deviation between the results of the primary and confirming tests,

$$100 \frac{q_{tc} - q_t}{q_t},$$

shall not be greater than plus or minus three (±3) per cent.

5.34 The correction factor, f, for deviation from *A.S.R.E. Standard Rating Conditions* of the test values of the saturated temperature of the leaving refrigerant vapor and the entering and leaving water or brine temperature shall be determined as follows:

$$f = \frac{t_e \text{ std} - t_l \text{ std}}{t_e \text{ test} - t_l \text{ test}} \times \frac{\log \left[\dfrac{t_e \text{ test} - t'_r \text{ test}}{t_l \text{ test} - t'_r \text{ test}}\right]}{\log \left[\dfrac{t_e \text{ std} - t'_r \text{ std}}{t_l \text{ std} - t'_r \text{ std}}\right]}$$

where t_e = temperature of entering water or brine, °F.

t_l = temperature of leaving water or brine, °F.

t'_r = saturated temperature of leaving refrigerant vapor, °F.

5.35 *A.S.R.E. Standard Ratings* shall be determined from test results as follows:

a. The *Net Refrigeration Effect*
$q_n = f\, q_w$ in Btu. per hr. or in tons.

b. The saturated temperature of the leaving refrigerant vapor shall be the standard rating value.

c. The flow rate of the water or brine in gal. per min. shall be the test value.

d. The water or brine pressure drop through the cooler, in lb. per sq. in., shall be the test value.

A.S.R.E. Circular No. 25-44

ASRE STANDARD
METHODS OF RATING AND TESTING FORCED CIRCULATION AND NATURAL CONVECTION AIR COOLERS FOR REFRIGERATION

by
THE JOINT COMMITTEE
ON
RATING COMMERCIAL REFRIGERATING EQUIPMENT

Sponsored by The American Society of Refrigerating Engineers
APPROVED
by ASRE
December, 1945

The American Society of Refrigerating Engineers
WEST 40th STREET
NEW YORK 18, N. Y.

Price 75 cents

[redacted]

Joint Committee on Rating Commercial Refrigerating Equipment

Glenn Muffly, Springfield, Ohio, *General Chairman*
Louis S. Morse, York, Pennsylvania, *Secretary*

For The American Society of Refrigerating Engineers (Sponsor)—
Glenn Muffly, *Chairman*
R. H. Tull
Frank R. Zumbro

For the Air Conditioning & Refrigerating Machinery Association—
Donald French, Chairman
Louis S. Morse

For the American Society of Heating & Ventilating Engineers—
F. H. Faust, Chairman

For the Refrigeration Equipment Manufacturers Association—
C. E. Scott
M. V. Stagg

For the Refrigeration Condensing Unit Manufacturers Association (Division of REMA)—
S. R. Hirsch
Clyde E. Ploeger

Special Sub-Committee:
R. D. Blum
C. E. Ehrenhardt
D. D. Wile

Acknowledgment is made of most valuable assistance by members of the St. Louis Section, A.S.R.E., Code 25 Revision Committee:
A. G. Loeffel, *Chairman*
B. R. Davidson
H. J. Kipp
R. C. Yantis

A. S. R. E. Standards Committee

B. H. Jennings, *Chairman*
Glenn Muffly
R. T. Brizzolara
S. R. Hirsch
J. G. Bergdoll, Jr.

Proposed A.S.R.E. Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration

SECTIONS

1. Scope and Purposes
2. Classification
3. Method of Rating
4. Instruments
5. Apparatus
6. Method of Testing
7. Test Procedure
8. Data to be Recorded
9. Computations
10. Definitions.

Section 1. SCOPE & PURPOSES

0 *Scope*

1.11 This standard prescribes methods of rating and testing forced circulation and natural convection air coolers for refrigeration. It does not include air conditioning units, for which see A.S.R.E. Circulars 13 and 16.

1.12 This standard applies to forced circulation and natural convection air coolers which operate under either frosting or non-frosting conditions.

1.13 It does not cover coolers which must be manually or automatically defrosted after running periods shorter than the three-hour test period specified in paragraph 3.42.

0 *Purposes*

1.21 The purposes of this standard are:
- a. To list and define the terms used in rating.
- b. To specify the data required and to establish conditions and methods of test for obtaining Standard Ratings.
- c. To establish methods of test which may be used and to specify the type of data which should be included in application ratings.

1.22 These standard tests are for rating only. They are not for production testing.

Section 2. CLASSIFICATION

Forced Circulation Air Coolers are classified as to type as follows:
- a. Dry coil
- b. Sprayed coil
- c. Spray—no coil

Forced Circulation Air Coolers are classified as to methods of discharging air to the room as follows:
- a. Free delivery fans
- b. Pressure fans

Natural Convection Air Coolers are classified as to type as follows:
- a. External baffle
- b. Built-in baffle
- c. Without baffle

Section 3. METHOD OF RATING

Ratings

3.11 In order to fulfill the requirements of this standard, two simultaneous methods of checking the ratings shall be used. One method shall measure the cooling capacity on the air side, the other shall measure the cooling capacity on the refrigerant side.

Table 1. Standard Rating Conditions

Group No.	*Dry bulb Temp., °F.*	*Relative humidity, %*	*Wet bulb depression, °F*	*Dew point depression, °F*	*Refrigerant Temp., °F.*
I	45	85	2.1	4.2	30
II	35	85	1.6	4.0	20
III	15	85	0.8	3.5	0
IV	0	85	0.5	3.0	—15

3.12 The Standard Rating shall be given under at least one of the group conditions specified in Table 1. Ratings of each type of air cooler shall be based on actual tests as follows:
- a. Forced circulation air coolers shall be tested with refrigerant temperatures and entering air conditions as specified in Table 1.
- b. Natural convection air coolers shall be tested by the calibrated box method (paragraph 5.50) with refrigerant temperature and average room air conditions as specified in Table 1.

3.13 Standard Ratings shall be at the barometric pressure of 29.92 in. Hg. For barometric pressures other than 29.92 in. Hg. dry and wet bulb temperatures shall be maintained as specified in Table 1; the dew point or relative humidity will then differ from Table 1 in accordance with known psychrometric data. For forced circulation air coolers, no change shall be made in fan speed or system resistance to correct for variations of barometer, but the air quantity shall be corrected to the standard pressure. If desired, the heat transfer rate can be corrected by increasing the test results 0.8 per cent for each inch of barometer reading below 29.92 at which the test was conducted.

3.14 The ratings are for continuous operation with no correction for the time required for defrosting.

3.15 Forced circulation pressure type air coolers shall be rated:
- a. On the gross cooling effect without the deduction for heat equivalent of motors or fan horsepower since these will vary with the installation conditions and are usually considered as part of the room load.
- b. With maximum recommended air volume when operating against ¼-in. water external static resistance.

3.16 Forced circulation free delivery air coolers shall be rated:

3.16
a. On their net cooling capacity, thus including the heat equivalent of the motor power since the motors form an integral part of the cooler and in use deduct from the cooling capacity.
b. With zero static pressure at the air discharge.

3.17 Cooling with Volatile Refrigerant. The temperature of the refrigerant in the air cooler shall be per Table 1 and is taken as the saturation temperature corresponding to the pressure at the outlet connection of the air cooler. The temperature of the liquid refrigerant entering the liquid control device or the heat interchanger, if one is supplied as standard equipment on the air cooler being tested, shall be 90°F. and its pressure shall not exceed that corresponding to 100°F. The air cooler must be operated so that liquid refrigerant does not flood out through the suction connection.

3.18 Cooling with Non-Volatile Refrigerant. The refrigerating medium shall enter the air cooler at a temperature per Table 1 and shall flow at the rate intended in the normal operation.

3.20 *Data Required in Standard Ratings*

3.21 The rating of an air cooler shall include all of the following data which apply to its function.

3.22 Data on Cooling shall include:
a. Gross cooling effect for forced circulation pressure type air coolers and net cooling effect for forced circulation free delivery air coolers.
b. Sensible heat ratio, sensible cooling effect divided by total cooling effect. (Optional for forced circulation free delivery and natural convection air coolers.)
c. Name of refrigerant or refrigerating medium for either volatile or non-volatile refrigerant. If a brine is used, its composition, density and specific heat should be stated.
d. Temperature and pressure of volatile refrigerant vapor leaving the air cooler.
e. Temperature and pressure of volatile refrigerant liquid entering the refrigerant control device or heat interchanger, if supplied.
f. Temperatures of non-volatile refrigerant entering and leaving the air cooler.
g. Quantity of non-volatile refrigerant circulated in gals. per min.
h. Pressure drop of non-volatile refrigerating medium in lb. per sq. in., between the inlet and outlet connections of the air cooler for the stated flow.
i. Power consumption of each auxiliary, such as pumps, where they are regularly supplied as part of the air cooler.
j. Spray pressure in lb. per sq. in. gage and the quantity of liquid sprayed in gal. per min.

3.23 Data on Air Flow shall include:
a. Air volume delivered by the air cooler in c.f.m of Standard Air as defined herein for the conditions specified in 3.15b. (Not required for natural convection air coolers.)
b. The value of external resistance in inches of water for pressure type air coolers, at rating conditions.
c. The value of internal resistance, in inches of water for pressure type air coolers, for which the air flow is specified.
d. R.p.m. of fan.
e. Fan motor power input, in watts for air coolers supplied with motor.
f. Fan horsepower required for air coolers not supplied with motor.
g. Horsepower (nameplate
tor, or motors of free delivery air coolers.

3.30 *Application Ratings*

3.31 It is recommended that application ratings include the following:
a. Total cooling effect for forced circulation pressure type air coolers and net cooling effect for forced circulation free delivery air coolers.
b. Sensible heat ratio.
c. Name of refrigerant.
d. Refrigerant temperature.
e. Quantity of non-volatile refrigerant circulated per minute.
f. Pressure drop through the coil when using non-volatile refrigerant.
g. Spray pressure.
h. Entering air dry bulb temperature, °F.
i. Entering air relative humidity, per cent.
j. Leaving air dry bulb temperature, °F.
k. Leaving air relative humidity, per cent.
l. Running period between defrostings.
m. Air quantity, c.f.m. of Standard Air.
n. Fan and spray pump motor horsepower.
o. Fan motor watts input for air coolers supplied with motors.
p. Pressure available for resistance of ducts in inches of water (pressure type air coolers only).

3.40 *Effect of Frost Formation*

3.41 For tests under the condition of Group I fo
any type, and for Groups II, III and IV wit
brine spray, there will be no frost. Unde
these conditions, each test need be run onl
long enough to establish equilibrium.

3.42 It is recognized that Groups II, III and IV wit
dry surface air coolers will result in fro
formation. Some expedient must, therefore, b
used to assure that the tests will show a pe
formance corresponding to a frost accumul
tion with reasonable frequency of defrostin
The rating for Group II shall be based on t
terminal conditions after 3 hr. of continuo
operation subsequent to defrosting. The ra
ings for Groups III and IV shall be based
the terminal condition at the end of 8 hr.
continuous operation subsequent to defrosti
except if the cooler accumulates excessive fro
the terminal condition may be taken after
period not less than three hours of continuo
operation and the duration of the run shall
stated. The terminal condition (that is the
erage of the last two sets of readings) sh
be taken as the basis of computing the air co
er performance. The weight of the mel
frost shall be determined at the end of the t
and shall be recorded by weight.

3.50 *Rating Tolerance*

3.51 In view of inequalities in manufacture, a va
tion of ± 5 per cent between the performance
a given air cooler and the rating of that air c
er will not be considered excessive.

3.60 *Published Ratings*

3.61 Published ratings shall include one or m
Standard Ratings identified by group num

Section 4. INSTRUMENTS

4.10 *Temperature Measuring Instruments*

4.11 Temperature measurements shall be made
one or more of the following instruments:
a. Mercury-in-glass thermometers.
b. Thermocouples.
c. Electric resistance thermometers.
d. Dew point instruments of Regnault or equ
lent type.

4.12 Accuracy shall be within the following limits:
- a. Wet and dry bulb and dew point temperatures, ± 0.1°F.
- b. Non-volatile refrigerant temperatures, ± 0.1°F.
- c. All other temperatures, ± 0.5°F.

4.13 In no case shall the smallest scale division of the measuring instrument exceed two times the specified accuracy.

4.14 Where an accuracy closer than 0.5°F. is specified, the instrument shall be calibrated by comparison to a certified standard in the range of use or shall itself be certified as to accuracy.

20 *Pressure Measuring Instruments for Refrigerant*

4.21 Pressure measurements shall be made with one or more of the following instruments:
- a. Mercury column.
- b. Bourdon tube gage.

4.22 Accuracy shall be within the following limits:
- a. Suction pressure, ± 0.5 per cent of the absolute pressure.
- b. Other pressures, ± 2 per cent of the absolute pressure.

4.23 Calibration of the Bourdon tube gage shall be with respect to a dead weight tester or by comparison with a mercury column.

4.24 In no case shall the smallest scale division of the measuring instrument exceed two and one half times the specified accuracy.

30 *Air Capacity Measuring Instruments*

4.31 Air capacity measurements shall be made with a standard nozzle used in conjunction with a manometer (draft gage) to be read to within 1 per cent of the head, or shall be made by Pitot tube if the installation is in accordance with 5.32 and 5.33.

4.32 Nozzles constructed in accordance with Fig. 1 are recommended and when installed in accordance with instructions given in Section 5 may be used without calibration with the discharge coefficient of 0.99 given in paragraph 9.11C.

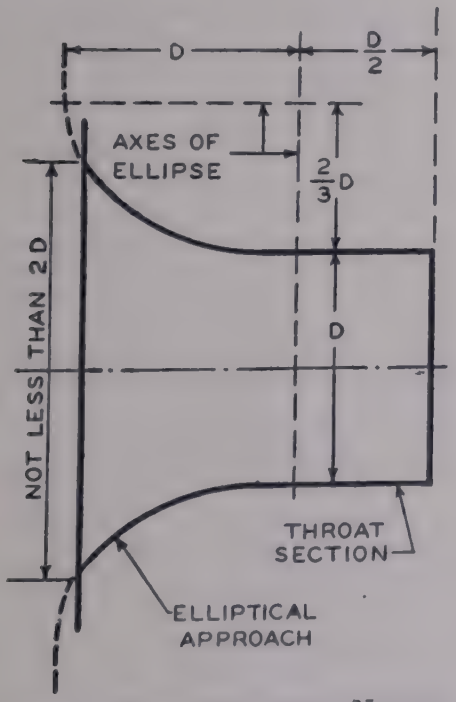


Fig. 1. Flow Measuring Nozzle

4.33 Where other type nozzles are employed, they shall be calibrated after installation by the steam condensate method as described in the latest edition of the A.S.H.V.E. Code for Testing and Rating Steam Unit Heaters.

4.34 Pitot tubes shall be of the accepted commercial type. A detailed description of the Pitot tube is given in the A.S.M.E. Test Code for Fans, Publication No. 11.

4.40 *Electrical Measuring Instruments*

4.41 Electrical measurements shall be made with one or more of the following instruments:
- a. Indicating.
- b. Integrating.

4.42 Accuracy shall be within the following limits:
- a. Instruments used for measuring the electrical input to heaters furnishing sensible or latent heat loads shall be accurate within 1 per cent of the quantity measured.
- b. Instruments used for measuring the electrical input to fan motors, pump motors or other accessories shall be accurate within 2 per cent of the quantity measured.

4.50 *Volatile Refrigerant Flow Measuring Instruments*

4.51 Flow measurements shall be made with one or more of the following instruments, having an accuracy within ± 2 per cent of the quantity measured:
- a. Liquid refrigerant quantity meter measuring either weight or volume.
- b. Liquid refrigerant flow meter.
- c. Calibrated suction line orifice.
- d. Condenser water method as defined in A.S.R.E. Circular #14.
- e. Calibrated condensing unit as defined in A.S.R.E. Circular #14.

4.60 *Brine or Water Flow Measuring Instruments*

4.61 Volume measurements shall be made with one or more of the following instruments having an accuracy within ± 2 per cent of the quantity measured:
- a. Liquid quantity meter, measuring either weight or volume.
- b. Liquid flowmeter.

4.70 *Speed Measuring Instruments*

4.71 Speed measurements shall be made with one or more of the following instruments having an accuracy within ± 1 per cent:
- a. Revolution counter.
- b. Tachometer.
- c. Stroboscope.
- d. Oscillograph.

4.80 *Specific Gravity Measuring Instruments*

4.81 Brine specific gravity shall be read with a hydrometer or other means accurate to 1 per cent of the specific gravity read.

4.90 *Other Measurements*

4.91 Time measurements shall be made with apparatus whose accuracy is within ± 0.2 per cent.

4.92 Weight measurement shall be made with apparatus whose accuracy is within ±0.2 per cent.

Section 5. APPARATUS

5.10 *Receiving Chamber with Single Nozzle*

5.11 The principle of this method is to measure the quantity and the change in properties of air entering and leaving an air cooler. The air quantity is determined from the velocity pressure at the discharge of a nozzle placed between a receiving and a discharge chamber. The change in air properties is determined from the temperature and moisture content at the inlet to the air cooler and at the discharge from the nozzle. (See Fig. 2).

5.12 The outlet or outlets of the air cooler shall be joined to a receiving chamber with or without a communicating duct. The duct and the receiving chamber shall be sealed so that air leakage at any static pressure within the range of

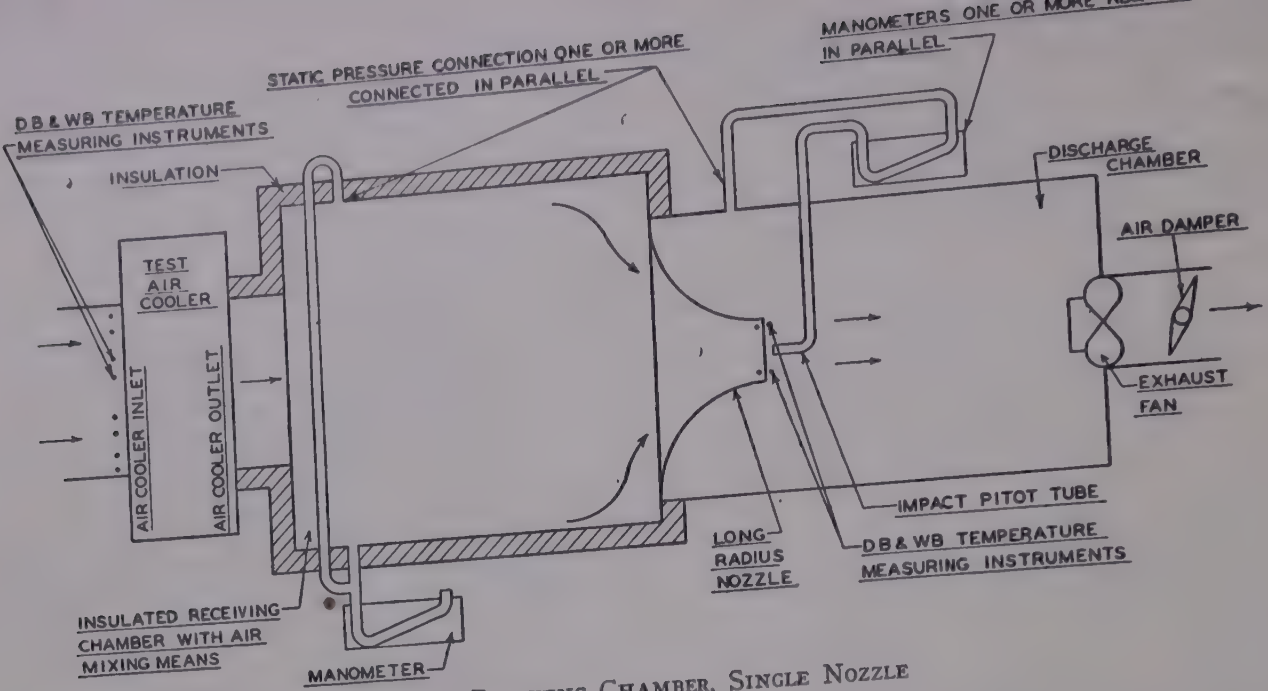

MANOMETERS ONE OR MORE
IN PARALLEL
STATIC PRESSURE CONNECTION ONE OR MORE CONNECTED IN PARALLEL
DB & WB TEMPERATURE MEASURING INSTRUMENTS
INSULATION
TEST AIR COOLER
AIR COOLER INLET
AIR COOLER OUTLET
DISCHARGE CHAMBER
AIR DAMPER
EXHAUST FAN
IMPACT PITOT TUBE
DB & WB TEMPERATURE MEASURING INSTRUMENTS
LONG RADIUS NOZZLE
INSULATED RECEIVING CHAMBER WITH AIR MIXING MEANS
MANOMETER

FIG. 2. RECEIVING CHAMBER, SINGLE NOZZLE

5.12 the test shall not exceed 1 per cent of the total air flow. The duct and the receiving chamber shall be insulated so that heat leakage shall not exceed 5 per cent of the capacity of the air cooler being tested. The heat leakage shall be calculated from data given in the *Refrigerating Data Book* and a correction to the air cooler capacity made accordingly. The ambient temperature used in this calculation shall be determined by not less than two temperature measuring instruments.

5.13 A nozzle constructed in accordance with Section 4.32 (Fig. 1) shall be fitted into one wall of the receiving chamber discharging into the discharge chamber. The diameter (dimension D) of the nozzle shall be such that the air velocity at the throat is not less than 2,000 ft. per min.

5.14 The size and arrangement of the receiving chamber shall be such as to provide uniform approach velocity to the nozzle. In no case shall the minimum dimension of the wall in which the nozzle is located be less than 2 throat diameters. The uniformity of the approach velocity shall be determined by a Pitot tube traverse and the maximum approach velocity shall not exceed the minimum approach velocity by more than 20 per cent. Suitable diffusion baffles may be used if desired.

5.15 One or more static pressure connections shall be located in the receiving chamber, not in the direct blast, and connected by air-tight tubes to a manometer.

5.16 An exhaust fan shall be connected to the discharge chamber if necessary to overcome the resistance of the receiving chamber, the discharge chamber and the nozzle in excess of the external resistance specified.

5.17 One or more manometers connected in parallel shall be provided for measuring pressures at the nozzle. One side of each manometer shall be connected to one or more static pressure connections located in the side walls of the discharge chamber and near the wall in which the nozzle is located. The other side of the manometer shall be connected to an impact tube arranged to measure the velocity pressure of the air at the nozzle outlet.

5.20 *Mixing Box with Multiple Nozzles*

5.21 The principle of this method is substantially the same as that of the receiving chamber with single nozzle except that more than one nozzle may be used and the temperature and humidity of the air leaving the air cooler are measured at the discharge of a mixing box located between the discharge of the air cooler and the receiving chamber. (See Fig. 3.)

5.22 The outlet or outlets of the air cooler shall be connected directly to a mixing box at the outlet of which temperature measurements are made in openings so restricted as to obtain at least 500 ft. per min. air velocity. The mixing box shall be insulated so that the heat leakage shall not exceed 5 per cent of the air cooler capacity. The heat leakage shall be calculated from data in the *Refrigerating Data Book* and correction of the air cooler capacity made accordingly. The ambient temperature used in this calculation shall be determined by not less than two temperature measuring instruments. One or more static pressure connections shall be provided in the side walls of this mixing box adjacent to the discharge connection of the air cooler for determining the static pressure at the discharge of the unit. Pressure connections shall be connected by means of air-tight tubes to a manometer. This mixing box shall be connected to one wall of the receiving chamber.

5.23 The receiving chamber shall be of sufficient size to provide uniform approach velocity to the nozzle or nozzles or shall have suitable diffusion baffles to accomplish this purpose. The area of the wall of the receiving chamber perpendicular to the axis of flow to the nozzles shall not be less than 12 times the area of the nozzles installed. If desired, the receiving chamber may be provided with a well-gasketed door.

5.24 One or more nozzles shall be fitted into one wall of the receiving chamber discharging into a discharge chamber and shall be of such a size that the throat velocity is not less than 2,000 ft. per min. Center to center distances between nozzles in use shall not be less than 3 throat diameters and the distance from the center of any nozzle to any of the four adjacent side walls shall not be less than 1.5 throat diameters. If the nozzles are of different diameters, the distance between axes shall be based upon the average diameters.

5.25 Nozzles installed in accordance with paragrap 5.24 can be considered to have a negligible correction for an approach velocity. The possibility of any of the discharge air entering directly into a nozzle shall be minimized by th use of diffusion baffles or by locating the nozzles in a wall adjacent to the wall where the air enters the chamber and at a different level so that the maximum approach peak velocity shall not exceed 10 per cent of the nozzle velocity. The maximum approach peak velocity is defined as the maximum approach velocity at the wall in which the nozzle is located as determined by a Pitot tube or other air measuring instruments.

5.26 The size and arrangement of the discharge chamber shall be such that the distance from the center of any nozzle to the adjacent side wall shall not be less than 1.5 throat diameters. The distance from the nozzles to the nearest obstruction shall not be less than 10 throat diameters unless suitable diffusion baffles are used.

5.27 An exhaust fan shall be connected to the discharge chamber and shall be of such capacity that it will overcome the resistance of the chambers, nozzle and mixing box. It shall be able to provide a zero static pressure at the outlet of the air cooler.

5.28 To measure the pressure drop across the nozzle, one or more manometers in parallel shall have one side connected to one or more static pressure connections located flush with the inner wall of the receiving chamber. The other side of the manometers shall be connected in a similar manner to one or more static pressure connections in the wall of the nozzle discharge chamber. The static pressure connections shall be located so as not to be affected by the air flow.

5.29 Temperature readings at the nozzle shall be used for air flow calculations.

5.30 *Pitot Tube Apparatus*

5.31 This method determines the heat removing capacity of an air cooler by measuring the properties of the entering and leaving air and the quantity of air. The air quantity is determined by measuring with a Pitot tube the velocity in a discharge duct.

5.32 The outlet or outlets of the air cooler shall be joined to a duct of such cross-sectional area as to maintain an air velocity of not less than 2,000 ft. per min. The duct shall be of sufficient length to overcome irregular flow that may exist at the outlet of the unit or as a result of the transformation piece connecting the duct to the air cooler. A detailed arrangement of the duct is shown in Fig. 4.

5.33 Pitot tube readings shall be taken at not less than 20 locations in a round or rectangular duct and not less than 16 locations in a square duct. These locations shall be at the center of equal areas as indicated in Fig. 4. Perforated plates or wire screens may be used and are recommended to even the air flow. Where the air leaving the air cooler has a rotational component, a straightening device must be used to correct this condition.

5.34 For free delivery type units, a fan shall be provided at the discharge of the duct for overcoming the duct resistance.

5.35 The mean velocity through the duct shall be the average of the velocities at each station. A convenient method of calculation is to take the square roots of the individual velocity pressure readings, then average these square roots and square this average to obtain the mean velocity pressure.

5.36 Static pressure connections shall be provided in a section of straight duct connecting the air cooler to the transformation piece as shown in Fig. 4 and shall be connected to a manometer for measuring the static pressure at the discharge of the air cooler.

5.40 *Calibrated Box for Free Delivery Forced Circulation Air Coolers*

5.41 This method determines the heat removing capacity of an air cooler by measuring the heat input to an enclosing box, the heat leakage of which is known. It is intended for natural convection and optionally for free delivery, forced circulation air coolers.

5.42 In no case shall the heat leakage from the box exceed 20 per cent of the gross cooling effect of the equipment being tested. The box must be sealed against infiltration of air and shall be fitted with a sealed door having insulation to match the walls. The calibrated box shall be installed in a test enclosure of a size which will provide clearances of not less than 24 in.

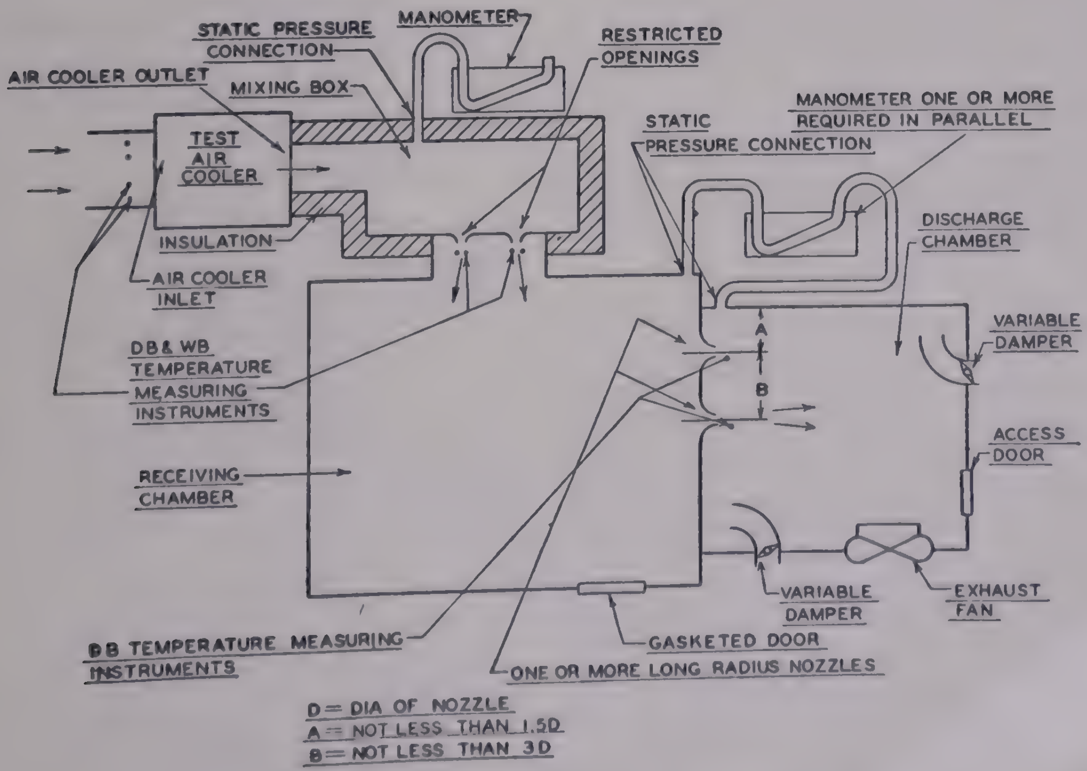

STATIC PRESSURE CONNECTION
MANOMETER
RESTRICTED OPENINGS
AIR COOLER OUTLET
MIXING BOX
TEST AIR COOLER
MANOMETER ONE OR MORE REQUIRED IN PARALLEL
STATIC PRESSURE CONNECTION
INSULATION
DISCHARGE CHAMBER
AIR COOLER INLET
VARIABLE DAMPER
A
DB& WB TEMPERATURE MEASURING INSTRUMENTS
B
ACCESS DOOR
RECEIVING CHAMBER
VARIABLE DAMPER
EXHAUST FAN
GASKETED DOOR
DB TEMPERATURE MEASURING INSTRUMENTS
ONE OR MORE LONG RADIUS NOZZLES
D= DIA OF NOZZLE
A= NOT LESS THAN 1.5D
B= NOT LESS THAN 3D

Fig. 3. Mixing Box, Multiple Nozzles

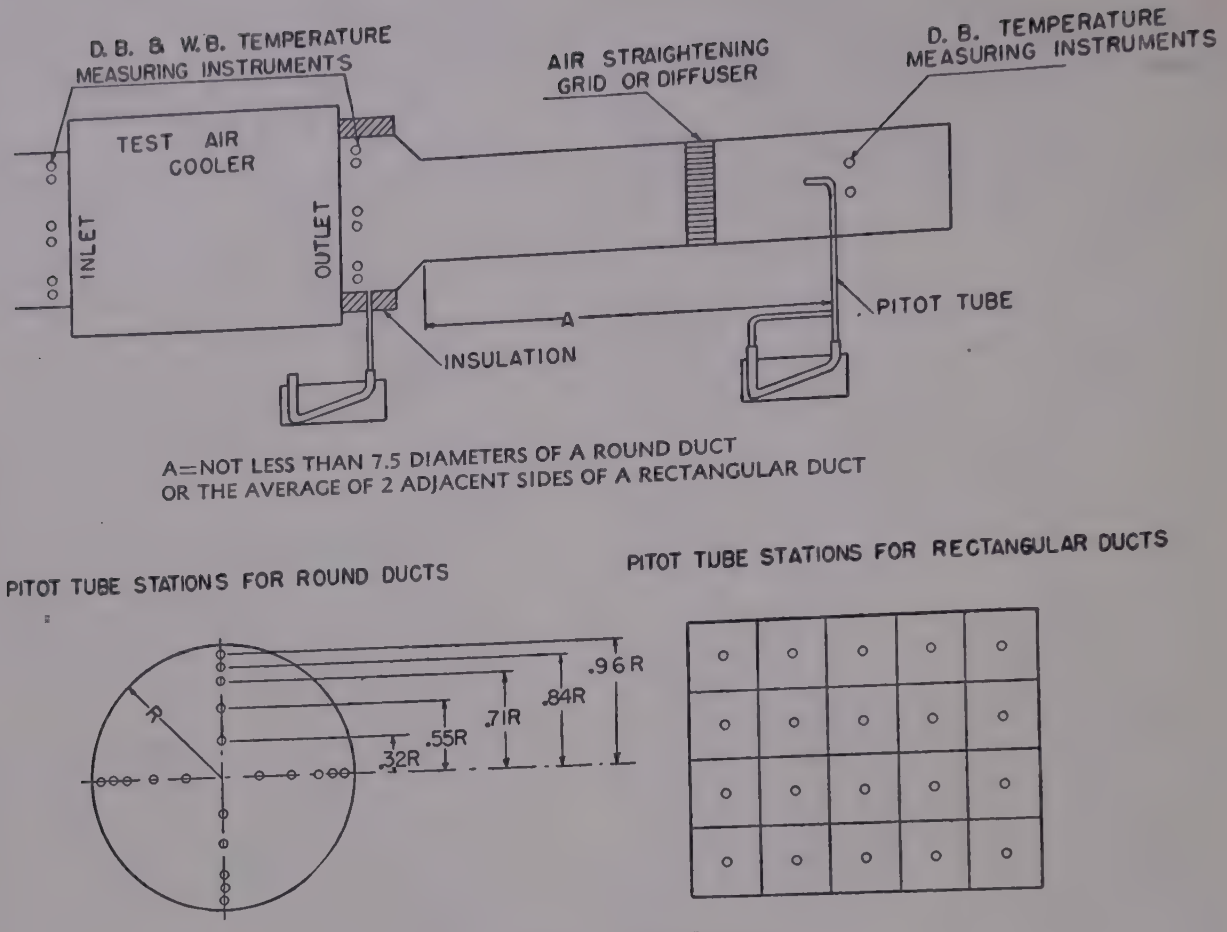

D. B. & W.B. TEMPERATURE MEASURING INSTRUMENTS
AIR STRAIGHTENING GRID OR DIFFUSER
D. B. TEMPERATURE MEASURING INSTRUMENTS
TEST AIR COOLER
INLET
OUTLET
PITOT TUBE
A
INSULATION
A=NOT LESS THAN 7.5 DIAMETERS OF A ROUND DUCT OR THE AVERAGE OF 2 ADJACENT SIDES OF A RECTANGULAR DUCT
PITOT TUBE STATIONS FOR ROUND DUCTS
PITOT TUBE STATIONS FOR RECTANGULAR DUCTS
R
.96R
.84R
.71R
.55R
.32R

FIG. 4. PITOT TUBE METHOD

5.42 at the sides and top and 3 in. under the floor. The size of the box shall be such that it shall not restrict the normal flow of air around the equipment to be tested unless the equipment is designed for use in an equally confined space.

5.43 Strip heaters or lamps of the desired capacity shall be installed and shielded as shown in Fig. 5 and shall be well distributed over the floor of the box. Power input to such heaters shall be measured and controlled. A rheostat or variable transformer may be used for control where the input is measured by indicating meters or the heaters may be operated intermittently where the input is measured by an integrating type meter. With intermittent heater operation, the length of the off-cycle shall not exceed 1 per cent of the length of the test run during which the input is being measured unless only a portion of the heater load is cycled in which case the maximum length of the off-cycle may be proportionally greater. If indicating meters are used, the voltage supplied to the heaters shall not vary during the test period by more than 1 per cent.

5.44 Latent heat may be supplied by any method which will distribute the moisture uniformly in the test box and will permit determination within ± 2 per cent of the total heat input to the humidifying means. It should be noted that the latent heat capacity of the air cooler is determined from the entering and leaving air properties and is not determined from the latent heat supplied to the test box, but the heat input to the humidifying means becomes a part of the total heat input to the test box. A suitable arrangement is a container filled with water in which is placed an immersion heater supplied by a controlled and measured electrical input.

5.45 Where thermometers are used, observation windows shall be provided to permit readings without the necessity of entering the test box or moving the thermometers.

5.46 Dew point apparatus may be used where desired to determine the humidity of the air provided that the sampling device does not interfere with the normal air circulation of the air cooler.

5.47 Thermometers for measuring the temperature around the outside of the box shall be located at the center of each wall and the ceiling and at a distance of 6 inches from the wall.

5.48 For the purpose of determining heat leakage through the walls of the calibrated box the inside temperature shall be assumed to be the average between the dry bulb temperatures entering and leaving the air cooler.

5.50 *Calibrated Box for Natural Convection Air Coolers*

5.51 The method of tests shall be as in paragraph 5.41.

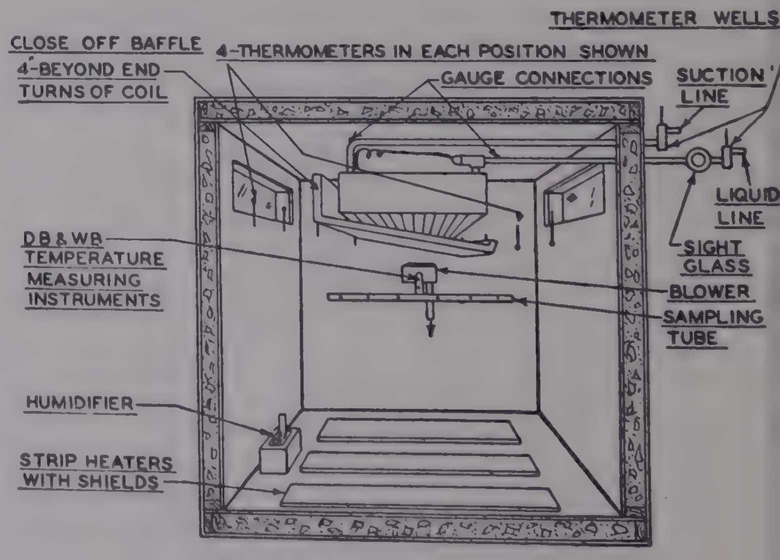

THERMOMETER WELLS
CLOSE OFF BAFFLE
4"-BEYOND END TURNS OF COIL
4-THERMOMETERS IN EACH POSITION SHOWN
GAUGE CONNECTIONS
SUCTION LINE
LIQUID LINE
SIGHT GLASS
DB & WB TEMPERATURE MEASURING INSTRUMENTS
BLOWER
SAMPLING TUBE
HUMIDIFIER
STRIP HEATERS WITH SHIELDS

FIG. 5. CALIBRATED BOX

5.52 The construction and location of the calibrated box shall be as in paragraph 5.42. The size shall be as in 5.42 except that dimensions shall be maintained as shown in Fig. 6 or Fig. 7.

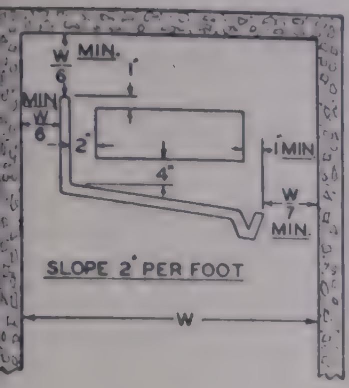


FIG. 6.
EXTERNAL BAFFLE TYPE

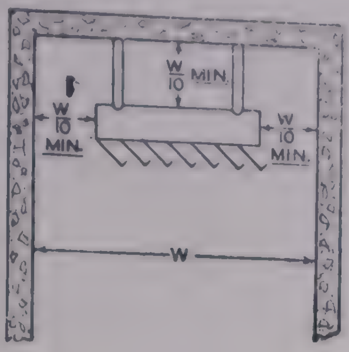


FIG. 7.
BUILT-IN BAFFLE TYPE

5.53 Heat shall be supplied to the calibrated box in accordance with paragraph 5.43 except that greater care must be exercised to prevent radiation to any part of the coil surface or the air measuring instruments and, furthermore, there shall be no uneven convection currents from the heaters.

5.54 Latent heat shall be supplied to maintain the wet bulb temperature as specified by any method which will permit determination within ± 2 per cent of the total heat input to the humidifying means and which will distribute the moisture uniformly in the test box. A suitable arrangement is specified in paragraph 5.44.

5.55 The average air temperature in the calibrated box shall be determined by eight pairs of wet and dry bulb measuring instruments, one of each located in each corner of the space below the baffle, spaced ¼ of the height of such space from floor and baffle, ¼ of the space width from each side and ¼ of the depth from front and rear walls. A wet and dry bulb measuring instrument may be located at a point in the calibrated box where the dry bulb temperature represents, within 0.5°F., the average of the eight dry bulb temperatures located as above. The dry bulb temperature shall be taken as average of the eight dry bulb instruments. An air sampling device may be substituted for the single pair of wet and dry bulb measuring instruments.

5.56 Observation windows shall be provided as in paragraph 5.45.

5.57 Dew point apparatus may be used as provided in paragraph 5.46.

5.58 Temperatures around the outside of the calibrated box shall be determined as in paragraph 5.47.

60 *Calibration of the Test Box* shall be performed by the heat leakage method. The calibration test shall be conducted as follows:

5.61 Five temperature measuring instruments shall be used to indicate the ambient temperature surrounding the test box. One instrument shall be placed in the center of each of the vertical walls and at the center of the ceiling and spaced six inches from the external surface on the test box. During calibration, the ambient temperature indicated by the average of the five readings shall be held constant to ± 2°F.

5.62 Eight temperature measuring instruments shall be used inside the test box to determine the average interior temperature. One of these instruments shall be located in each corner of the box, spaced ¼ of the height, ¼ of the width and ¼ of the depth of the box from each side. During calibration, the internal temperature shall be considered as the average of the eight temperature readings and shall be held constant to ±.0.5°F.

5.63 Heat leakage tests shall be made for the maximum and minimum forced air movements expected in the use of the test box. The calibration heat leakage shall then be plotted as a straight line function between these two air movements and this curve used as calibration for the test box. During each heat leakage test, the heater input shall be adjusted to maintain an average inside temperature approximately 25°F. above the outside temperature.

5.64 After thermal equilibrium has been established, the temperature shall be held constant as specified above and external and internal temperatures shall be recorded each hour for not less than ten hours.

5.65 This method is recommended but the test box may be calibrated for heat leakage by any other equivalent method.

5.70 *Volatile Refrigerant Measurements*

5.71 Pressure measuring instruments shall be placed in the liquid line on the high pressure side of the liquid control device, or the heat interchanger if one is supplied, and at the suction connection from the air cooler.

5.72 Temperature measuring instruments shall be placed so as to measure accurately the temperature of the liquid entering the liquid control device, or heat interchanger if one is supplied, and the vapor adjacent to the suction pressure gage connection. The lines shall be insulated at and adjacent to the instruments.

5.73 The liquid control device used during the test shall be the same as furnished or recommended for use with the air cooler.

5.74 Suction pressure at the air cooler shall be controlled by regulating the capacity of the refrigerant compressor. This may be accomplished by any desired means which will maintain steady state conditions. Suitable means are a hand-operated throttling valve in the suction or a hand-operated bypass valve between the high and low sides of the compressor.

5.80 *Non-Volatile Refrigerant Measurements*

5.81 Temperature measuring instruments shall be placed to measure accurately the temperature of the refrigerant entering and leaving the air cooler. The lines shall be insulated at and adjacent to the instruments.

Section 6. METHOD OF TESTING

6.10 *Basis of Rating*

6.11 In order to fulfill the requirements of this Standard, two simultaneous methods of checking the ratings shall be used. One method shall measure the cooling capacity on the air side, the other shall measure the cooling capacity on the refrigerant side.

6.12 The capacities measured on the air side and on the refrigerant side must agree within 6 per cent. The average of the two measurements shall be used for the rating of the unit.

6.20 *Refrigerant Side Test* shall consist of determination of cooling effect by one of the following methods:

6.21 Volatile refrigerant.

6.21

a. Condenser water method as defined in A.S.R. E. Circular 14.

b. Refrigerant weight or volume method as defined in A. S. R. E. Circular 14.

c. Refrigerant flow rate (liquid or gas) provided that the meter has been checked within 2 per cent of items 1 or 2 or a calorimeter within six months prior to the test.

d. Calibrated condensing unit method as defined in A.S.R.E. Circular 14 provided that prior to the test the unit is calibrated for the conditions under which it will operate during the test and that a recalibration of the condensing unit immediately after the test checks the original calibration within ± 2 per cent.

6.22 Non-volatile refrigerant. This test shall be made by determining the heat absorbed by the refrigerant and is obtained from the rate of flow, specific heat, and temperature rise of the refrigerant.

6.30 *Air Side Test* shall consist of determination of cooling effect by one of the following methods:

6.31 For pressure type forced circulation air coolers and optionally for free delivery forced circulation air coolers, the sensible and the total cooling effect shall be obtained from the air volume and the difference between properties of the air entering and leaving the air cooler. The air volume shall be directly measured by one of the following methods:

a. Receiving chamber with single nozzle.

b. Mixing box with multiple nozzles.

c. Pitot tube method.

6.32 For natural convection air coolers and optionally for free delivery forced circulation air coolers, the total cooling effect shall be obtained from the total heat input to a calibrated box. The sensible heat ratio shall be determined from the properties of the air entering and leaving the air cooler. The air quantity shall be calculated from the total heat input and the properties of the air entering and leaving the air cooler.

Section 7. TEST PROCEDURE

7.10 *Volatile Refrigerant Tests*

7.11 After establishing the entering air conditions, place the refrigerating system in operation and adjust the suction line throttling means to obtain the suction pressure corresponding to the refrigerant temperature in Table 1. The liquid control device shall then be adjusted to maintain the superheat at which the air cooler is designed to operate.

7.12 If the suction pressure or the suction temperature fluctuate due to cycling of the liquid control device, a continuous record shall be kept of the suction pressure and the suction temperature. The average temperature equivalent to the pressure shall not be permitted to deviate more than 0.5°F. and the maximum more than 3°F. from the refrigerant temperatures specified in Table 1. The suction temperature shall be taken as the average of the recorded readings. In no case, shall the test be acceptable if liquid refrigerant floods through at any time.

7.20 *Non-Volatile Refrigerant Tests*

7.21 After establishing the entering air conditions, adjust the non-volatile refrigerant temperature to the value in Table 1 and the flow rate as required. The refrigerant temperature shall not be permitted to vary more than 0.2°F. from the temperature specified. The refrigerant flow rate shall not be permitted to vary more than 2 per cent.

7.30 *Brine Concentration*

7.31 With spray type units, means shall be provided to maintain the density of the brine sufficiently constant to avoid noticeable effect upon the leaving air properties.

7.40 *Air Property Measurements*

7.41 Entering air shall be provided at the air conditions shown in Table 1. During any test run, the arithmetical average of all readings of entering air temperature at any point shall be held to that specified in Table 1 within 0.5°F. dry bulb, 0.3°F. wet bulb or 0.5°F. dew point. No individual dry bulb temperature reading of entering air shall vary by more than 2°F. from that specified in Table 1. No individual wet bulb temperature reading of entering air shall vary by more than 1°F. from that specified in Table 1. Wet bulb instruments shall be exposed to an air velocity of at least 300 ft. per min.

7.42 For dry type air coolers under Groups II, III and IV test conditions, there will be a progressive decrease in the air volume as frost accumulates and a corresponding change in the leaving air temperature and the static pressure. The test conditions for rating frost-forming air coolers are to be those at the end of the test run.

7.43 Special care is necessary to obtain reliable wet bulb readings at temperatures below freezing. Refer to Appendix "A" for information on this subject.

7.44 The method for obtaining entering and leaving air properties of forced circulation air coolers may be modified by the use of sampling tubes provided that they cause no detectable change of air property or velocity.

7.45 Dew point readings may be taken with apparatus of the Regnault type or other types of established accuracy. Dew point readings are taken when dew appears on the polished surface, not waiting for it to turn into frost if there is any detectable time difference. Suitable multiple-glass windows are required so that the dew point apparatus is visible with no outside temperature or humidity disturbance. An explanation of the Regnault type of dew point apparatus will be found in the U. S. Weather Bureau's publication W. B. No. 235 entitled "Psychrometric Tables."

7.46 Wet bulb and dry bulb air temperatures of the air entering and leaving the air cooler shall be taken at sufficient number of stations in the inlet and outlet flow areas to insure a true average. In no case shall there be less than one pair of stations per sq. ft. of flow area or less than a total of four stations.

7.50 *Air Flow Measurements*

7.51 The air flow of pressure type air coolers calculated from the test data shall not deviate from the air flow to be shown in the ratings by more than ± 5 per cent.

7.60 *Duration of Tests*

7.61 For tests under Group I and for tests of brin spray units under Groups II, III and IV ther will be no frost, and the test need be run onl long enough to establish equilibrium. The rat ing data shall be obtained from four consecutiv readings at 20-min. intervals which are withi the specified tolerances.

7 62 For Groups II, III and IV tests where frostin of the coil takes place, establish test condition then defrost as quickly as possible and immed ately re-establish test conditions at which tim the test run shall be considered to start. Th heater and dehumidifying means used to esta lish test conditions may be reduced or turne

7.62

off during the defrosting process in order to better maintain test conditions.

7.63 For the calibrated box method, dry bulb temperatures as specified for the test shall be maintained for at least 8 hr. prior to starting the test run. If desired, strip heater input and evaporation load may be reduced or turned off during this period and the condensing unit may be permitted to cycle automatically. At the end of this period, establish test conditions within the prescribed accuracy and start the test run. For Group II, III and IV tests, where the coils become frosted, it is permissible to defrost when necessary during the 8-hr. preliminary period. Before starting the test run, defrost the air cooler as quickly as possible and immediately establish test conditions.

7.64 For Groups II, III and IV tests where frost is formed, the duration of the test shall be as given in paragraph 3.42. The readings shall be recorded every 20 min. but the average of only the last two sets of readings shall be taken as the basis of computing the performance. The test readings shall be plotted to indicate that the last two sets of readings are consistent with previous readings within the established limit of error.

7.70 *General*

7.71 Fan speed shall be maintained within ± 2 per cent of rated speed.

7.72 Barometric pressure shall be read once during the period of the test during which readings for rating are taken.

Section 8. DATA TO BE RECORDED

8.10 *General Data*

a. Date.
b. Observers.
c. Barometric pressure.

8.20 *Data of Air Cooler*

a. Manufacturer's name and address.
b. Size, type and designation.
c. Speed of air cooler fan or fans.
d. Name of refrigerant or refrigerant medium. If brine is used, its composition, density and specific heat should be recorded.
e. Data for each motor:
    1. Where used.
    2. Rated h.p.
    3. Rated r.p.m.
    4. Manufacturer's name.
    5. Model or type.
    6. Rated voltage, phases and cycles.
f. Power consumption and applied voltage for each motor.
g. Temperature of air surrounding the air cooler.
h. Static pressure at discharge of cooling unit.
i. Static pressure at the inlet to the air cooler where ductwork or other obstruction is present.
j. For spray type units, record the spray pressure and quantity of brine or water sprayed.

30 *Air Side Test Data*

8:31 Receiving chamber with single nozzle—
a. Dry bulb and welt bulb or dew point temperatures of air entering the air cooler.
b. Dry bulb and wet bulb or dew point temperatures of air leaving the nozzle.
c. Temperature of air surrounding the receiving chamber.
d. The impact pressure at the discharge of the nozzle in the receiving chamber.
e. Diameter of the nozzle.

8.32 Mixing box with multiple nozzles—
a. Dry bulb and wet bulb or dew point temperatures of air entering air cooler.
b. Dry bulb and wet bulb or dew point temperatures of air leaving mixing box.
c. Temperature of air surrounding mixing box.
d. Dry bulb temperature of air leaving the air measuring nozzles.
e. The static pressure difference across the air measuring nozzles.
f. The number and diameters of nozzles used.

8.33 Pitot tube method—
a. Dry bulb and wet bulb or dew point temperatures of air entering the air cooler.
b. Dry bulb and wet bulb or dew point temperatures of air leaving the air cooler.
c. Dry bulb temperature of air at the point of Pitot tube readings.
d. Velocity pressure difference between the impact and static connection of the Pitot tube.

8.34 Calibrated box for free delivery forced circulation air coolers—
a. Dry bulb temperature entering the air cooler.
b. Wet bulb or dew point temperature entering the air cooler.
c. Dry bulb temperature leaving the air cooler.
d. Wet bulb or dew point temperature leaving the air cooler.
e. Dry bulb temperature outside the test box.
f. Electrical power input to sensible heat elements.
g. Electrical power input to humidifier.
h. Electrical power input to all motors and auxiliaries.

8.35 Calibrated box for natural convection air coolers.
a. Dry bulb temperature inside the box.
b. Wet bulb or dew point temperature inside the box.
c. Dry bulb temperature outside the box.
d. Electrical power input to heating elements.
e. Electrical power input to humidifier.
f. Electrical power input to all motors and auxiliaries.

8.40 *Refrigerant Side Test Data*

8.41 Volatile refrigerant.
a. Temperature and pressure of volatile refrigerant vapor leaving the air cooler.
b. Temperature and pressure of volatile refrigerant liquid entering the refrigerant control device, or heat interchanger if supplied.
c. Other data required for determining refrigerant side capacity shall be recorded as stated in the A.S.R.E. Circular 14 for the particular test method used.

8.42 Non-volatile refrigerant.
a. Temperatures of the non-volatile refrigerant entering the air cooler.
b. Temperatures of non-volatile refrigerant leaving the air cooler.
c. Quantity of non-volatile refrigerant circulated.
d. Pressure drop of non-volatile refrigerant between the inlet and outlet connections of the air cooler.
e. Density and specific heat of non-volatile refrigerant.

Section 9. COMPUTATIONS

9.10 *Symbols*

9.11 Primary symbols:

A = Nozzle area or duct area, sq. ft.

C = Nozzle coefficient of discharge. The coefficient of discharge is not fixed for any particular nozzle but depends upon the nozzle diameter, air velocity, viscosity and density. For air within the normal barometric variations and for the temperature range specified in this standard and for

9.11

nozzles of 5 in. diameter and larger the coefficient of discharge can be expressed with sufficient accuracy as 0.99.

E = Total electrical input to motor or motors of forced circulation free delivery air coolers, watts.

E_c = Total electrical input to heaters and auxiliary equipment of calibrated box during rating test, watts.

E_k = Total electrical input to heaters and fans (if used) of calibrated box during calibration, watts.

H = Enthalpy (total heat) of the air-water vapor mixture, Btu. per lb. of dry air. For air within the normal barometric variations and for the temperature range specified in this standard the enthalpy (total heat) of air-water vapor mixtures can be expressed with sufficient accuracy as given in the *Refrigerating Data Book* for a barometric pressure of 29.92 in. of mercury.

h = Velocity pressure at nozzle throat or the static pressure difference across the nozzle or velocity pressure in duct, in. of water.

I = Enthalpy (total heat) of volatile refrigerant, Btu. per lb.

K_c = Heat leakage coefficient of calibrated box, Btu. per hr. per °F.

K_m = Heat leakage coefficient of mixing box, Btu. per hr. per °F.

K_r = Heat leakage coefficient of receiving chamber, Btu. per hr. per °F.

P = Power input to fan of forced circulation pressure type air coolers, brake h.p.

Q = Air flow rate, c.f.m.

q_c = Heat leakage of calibrated box, Btu. per hr.

q_l = Latent cooling capacity, Btu. per hr.

q_s = Sensible cooling capacity, Btu. per hr.

q_t = Total cooling capacity, Btu. per hr.

S = Specific heat of non-volatile refrigerant, Btu. per lb. per °F.

T = Temperature of non-volatile or volatile refrigerant, °F.

t = Dry bulb temperature of air, °F.

t' = Wet bulb temperature of air, °F.

v = Specific volume of air-water vapor mixture, cu. ft. per lb. dry air. For air within the normal barometric variations and for the temperature range specified in this standard the specific volume of air-water vapor mixture can be expressed with sufficient accuracy as given in the *Refrigerating Data Book* for a barometric pressure of 29.92 in. of mercury.

V = Air velocity at nozzle throat or duct velocity at point of Pitot tube traverse, ft. per min.

W = Flow rate of volatile or non-volatile refrigerant, lb. per hr.

9.12 Subscripts:

1 = Property of air or refrigerant entering the air cooler.

2 = Property of air or refrigerant leaving the air cooler.

3 = Average property of air within calibrated box

4 = Average property of air surrounding calibrated box.

5 = Average property of air surrounding mixing box.

6 = Average property of air surrounding receiving chamber.

a = Refers to air side computation.

g = Refers to gross cooling capacity.

h = Refers to net cooling capacity.

m = Property of air at the outlet of the mixing box.

n = Property of air at the air measuring nozzle or at point of Pitot tube traverse.

r = Refers to the standard rating computation.

x = Property of air corrected to standard barometric pressure of 29.92 in. of mercury and standard temperature of 70° = 0.075 lb. per cu. ft.

z = Refers to refrigerant side.

9.20 *Refrigerant side cooling capacity calculations*

9.21 Calculation of gross total cooling capacity is as follows:

a. Non-volatile refrigerant

$$q_{tgz} = WS\ (T_2 - T_1) \qquad (1)$$

where:

q_{tgz} = Gross total cooling capacity, Btu. per hr.

W = Weight flow of refrigerant, lb. per hr.

S = Specific heat of refrigerant, Btu. per lb. per °F.

T_1 = Temperature of refrigerant entering air cooler, °F.

T_2 = Temperature of refrigerant leaving air cooler, °F.

b. Volatile refrigerant

$$q_{tgz} = W\ (I_2 - I_1) \qquad (2)$$

where:

q_{tgz} = Gross total cooling capacity, Btu. per hr.

W = Weight flow of refrigerant, lbs. per hr

I_1 = Enthalpy (total heat) of the refrigerant liquid entering the air cooler, Btu per lb. of refrigerant.

I_2 = Enthalpy (total heat) of the refriger ant vapor leaving the air cooler, Btu per lb. of refrigerant.

9.22 Calculation of net total cooling capacity (re quired for forced convection free delivery ai coolers only) for volatile and non-volatil refrigerant.

$$q_{thz} = q_{tgz} - 3.41\ E \qquad (3$$

where:

q_{thz} = Net total cooling capacity, Btu. per h

q_{tgz} = Gross total cooling capacity, Btu. pe hr.

E = Electrical input to fan motor, watt

9.30 *Air flow calculations*

9.31 Calculations of air flow are as follows:

a. Nozzles method (for forced circulation a coolers only)

The air velocity through nozzles for eith receiving chamber with single nozzle or mi ing box with multiple nozzles is as follows:

$$V = 1096\ \sqrt{hv_n} \qquad ($$

where:

V = Nozzle velocity per min.

h = Velocity pressure at nozzle throat the static pressure difference acr nozzle, in. of water.

v_n = Specific volume of air-water va mixture at the nozzle, cu. ft. per dry air.

The air flow rate is as follows:

$$Q_n = CAV \tag{5}$$

where:

Q_n = Air flow rate at the nozzle, c.f.m.

C = Nozzle coefficient.

A = Nozzle area, sq. ft.

V = Nozzle velocity, ft. per min.

The air flow rate for standard conditions is as follows:

$$Q_s = \frac{Q_n}{v_n \times 0.075} \tag{6}$$

where:

Q_s = Air flow rate at standard conditions of 70°F. and 29.92 in. of mercury (0.075 lb. dry air per cu. ft.), c.f.m.

Q_n = Air flow rate at the nozzle, c.f.m.

v_n = Specific volume of air-water vapor mixture at the nozzle, cu. ft. per lb. of dry air.

b. Pitot tube traverse method (for forced circulation air coolers only)

The air velocity through the duct is as follows:

$$V = 1096 \sqrt{hv_n} \tag{7}$$

where:

V = Duct velocity at the point of Pitot tube traverse, ft. per min.

h = Velocity pressure in duct at the point of Pitot tube traverse, in. of water.

v_n = Specific volume of air-water vapor mixture in duct at the point of Pitot tube traverse, cu. ft. per lb. of dry air.

The air flow rate is as follows:

$$Q_n = AV \tag{8}$$

where:

Q_n = Air flow rate at point of Pitot tube traverse, c.f.m.

A = Duct area at point of Pitot tube traverse, sq. ft.

V = Duct velocity at the point of Pitot tube traverse, ft. per min.

The air flow rate for standard conditions is as follows:

$$Q_s = \frac{Q_n}{v_n \times 0.075} \tag{9}$$

where:

Q_s = Air flow rate at standard conditions of 70°F. and 29.92 in. of mercury (0.075 lb. dry air per cu. ft.), c.f.m.

Q_n = Air flow rate at point of Pitot tube traverse, c.f.m.

v_n = Specific volume of air-water vapor mixture at point of Pitot tube traverse, cu. ft. per lb. of dry air.

c. Calibrated box method (for forced circulation free delivery air coolers only)

The air flow rate is as follows:

$$Q_1 = \frac{q_{tna}\ v_1}{60\ (H_1 - H_2)} \tag{10}$$

where:

Q_1 = Air flow rate entering the air cooler, c.f.m.

q_{tna} = Net total cooling capacity, Btu. per hr.

v_1 = Specific volume of air-water vapor mixture entering the air cooler, cu. ft. per lb. of dry air.

H_1 = Enthalpy (total heat) of air-water vapor mixture entering the air cooler, Btu. per lb. of dry air.

H_2 = Enthalpy (total heat) of air-water vapor mixture leaving the air cooler, Btu. per lb. of dry air.

The air flow rate for standard conditions is as follows:

$$Q_s = \frac{Q_1}{v_1 \times 0.075} \tag{11}$$

where:

Q_s = Air flow rate at standard conditions of 70°F. and 29.92 in. of mercury (0.075 lb. of dry air per cu. ft.), c.f.m.

Q_1 = Air flow rate entering the air cooler, c.f.m.

v_1 = Specific volume of air-water vapor mixture entering the air cooler, cu. ft. per lb. of dry air.

9.40 *Heat leakage coefficient calculations*

9.41 The heat leakage coefficient in Btu. per hr. per °F. for the mixing box or the receiving chamber is calculated from the product of the square feet of exposed area of the box or chamber by the heat transmission coefficient for the insulation used, which data can be obtained from the *Refrigerating Data Book.*

9.42 Calculation of heat leakage coefficient of the calibrated box is as follows:

$$K_c = \frac{3.41\ E_k}{t_3 - t_4} \tag{12}$$

where:

K_c = Heat leakage coefficient of calibrated box, Btu. per hr. per °F.

E_k = Total electrical input to heaters and fans (if used) of calibrated box, watts.

t_3 = Average dry bulb temperature of air within calibrated box, °F.

t_4 = Average dry bulb temperature of air surrounding calibrated box, °F.

9.50 *Air side cooling capacity calculations*

9.51 Calculation of gross total, gross sensible and gross latent cooling capacities (required for forced convection pressure type and natural convection air coolers).

a. Nozzle method (for forced convection pressure type air coolers only)

1. Receiving chamber with single nozzle.

$$q_{tga} = \frac{60\ Q_n}{v_n} (H_1 - H_n) + K_r\ (t_6 - t_n) + 2544\ P \tag{13}$$

$$q_{sga} = \frac{60\ Q_n}{v_n} \times .24\ (t_1 - t_n) + K_r\ (t_6 - t_n) + 2544\ P \tag{14}$$

$$q_{lga} = q_{tga} - q_{sga} \tag{15}$$

2. Mixing box with multiple nozzles.

$$q_{tga} = \frac{60\ Q_n}{v_n} (H_1 - H_m) + K_m\ (t_5 - t_m) + 2544\ P \tag{16}$$

$$q_{sga} = \frac{60\ Q_n}{v_n} \times .24\ (t_1 - t_m) + K_m\ (t_5 - t_m) + 2544\ P \tag{17}$$

$$q_{lga} = q_{tga} - q_{sga} \tag{18}$$

where:

q_{tga} = Gross total cooling capacity, Btu. per hr.

q_{sga} = Gross sensible cooling capacity, Btu. per hr.

q_{lga} = Gross latent cooling capacity, Btu. per hr.

Q_n = Air flow rate at the nozzle, c.f.m.

v_n = Specific volume of air-water vapor mixture at the nozzle, cu. ft. per lb. of dry air.

H_1 = Enthalpy (total heat) of air-water vapor mixture entering the air cooler, Btu. per lb. of dry air.

H_m = Enthalpy (total heat) of air-water vapor mixture leaving the mixing box, Btu. per lb. of dry air.

H_n = Enthalpy (total heat) of air-water vapor mixture at the nozzle, Btu. per lb. of dry air.

K_m = Heat leakage coefficient of mixing box, Btu. per hr. per °F.

K_r = Heat leakage coefficient of receiving chamber, Btu. per hr. per °F.

P = Power input to fans, h.p.

t_1 = Dry bulb temperature of air entering the air cooler, °F.

t_m = Dry bulb temperature of air leaving the mixing box, °F.

t_n = Dry bulb temperature of air at the nozzle, °F.

t_5 = Average dry bulb temperature of air surrounding the mixing box, °F.

t_6 = Average dry bulb temperature of air surrounding the receiving chamber, °F.

b. Pitot tube traverse method (for forced convection pressure type air coolers only)

$$q_{tga} = \frac{60\,Q_n}{v_n}(H_1 - H_2) + 2544\,P \tag{19}$$

$$q_{sga} = \frac{60\,Q_n}{v_n} \times .24\,(t_1 - t_2) + 2544P \tag{20}$$

$$q_{lga} = q_{tga} - q_{sga} \tag{21}$$

where:

q_{tga} = Gross total cooling capacity, Btu. per hr.

q_{sga} = Gross sensible cooling capacity, Btu. per hr.

q_{lga} = Gross latent cooling capacity, Btu. per hr.

Q_n = Air flow rate at the point of Pitot tube traverse, c.f.m.

v_n = Specific volume of air-water vapor mixture at the point of Pitot tube traverse, cu. ft. per lb. of dry air.

H_1 = Enthalpy (total heat) of air-water vapor mixture entering the air cooler, Btu. per lb. of dry air.

H_2 = Enthalpy (total heat) of air-water vapor mixture leaving the air cooler, Btu. per lb. of dry air.

t_1 = Dry bulb temperature of air entering the air cooler, °F.

t_2 = Dry bulb temperature of air leaving the air cooler, °F.

P = Power input to fans, h.p.

c. Calibrated box method (for natural convection air coolers only)

$$q_{tga} = K_c\,(t_4 - t_3) + 3.41\,E_c \tag{22}$$

where:

q_{tga} = Gross total cooling capacity, Btu. per hr.

K_c = Heat leakage coefficient of calibrated box, Btu. per hr. per °F.

E_c = Total electrical input to heaters and auxiliary equipment of calibrated box, watts.

t_3 = Average dry bulb temperature of air within the calibrated box, °F.

t_4 = Average dry bulb temperature of air surrounding the calibrated box, °F.

9.52 Calculation of net total, net sensible and net latent cooling capacities (required for forced convection free delivery air coolers only)

a. Nozzle methods:

1. Receiving chamber with single nozzle.

$$q_{tha} = \frac{60\,Q_n}{v_n}(H_1 - H_n) + K_r\,(t_6 - t_n) \tag{23}$$

$$q_{sha} = \frac{60\,Q_n}{v_n} \times .24\,(t_1 - t_n) + K_r\,(t_6 - t_n) \tag{24}$$

$$q_{lha} = q_{tha} - q_{sha} \tag{25}$$

2. Mixing box with multiple nozzles.

$$q_{tha} = \frac{60\,Q_n}{v_n}(H_1 - H_m) + K_m\,(t_5 - t_m) \tag{26}$$

$$q_{sha} = \frac{60\,Q_n}{v_n} \times .24\,(t_1 - t_m) + K_m\,(t_5 - t_m) \tag{27}$$

$$q_{lha} = q_{tha} - q_{sha} \tag{28}$$

where:

q_{tha} = Net total cooling capacity, Btu. per hr

q_{sha} = Net sensible cooling capacity, Btu. pe hr.

q_{lha} = Net latent cooling capacity, Btu. pe hr.

Q_n = Air flow rate at the nozzle, c.f.m.

v_n = Specific volume of air-water vapo mixture at the nozzle, cu. ft. per lb of dry air.

H_1 = Enthalpy (total heat) of air-wate vapor mixture entering the air coole Btu. per lb. of dry air.

H_m = Enthalpy (total heat) of air-wate vapor mixture leaving the mixing bo Btu. per lb. of dry air.

H_n = Enthalpy (total heat) of air-wat vapor mixture at the nozzle, Btu. p lb. of dry air.

K_m = Heat leakage coefficient of mixing bo Btu. per hr. per °F.

K_r = Heat leakage coefficient of receivi chamber, Btu. per hr. per °F.

t_1 = Dry bulb temperature of air enter the air cooler, °F.

t_m = Dry bulb temperature of air leav the mixing box, °F.

t_n = Dry bulb temperature of air at nozzle, °F.

t_5 = Average dry bulb temperature of surrounding the mixing box, °F.

t_6 = Average dry bulb temperature of surrounding the receiving chaml °F.

b. Pitot tube traverse method.

$$q_{tha} = \frac{60\,Q_n}{v_n}(H_1 - H_2) \quad ($$

$$q_{sha} = \frac{60\ Q_n}{v_n} \times .24\ (t_1 - t_2) \qquad (30)$$

$$q_{lha} = q_{tha} - q_{sha} \qquad (31)$$

where:

q_{tha} = Net total cooling capacity, Btu. per hr.
q_{sha} = Net sensible cooling capacity, Btu. per hr.
q_{lha} = Net latent cooling capacity, Btu. per hr.
Q_n = Air flow rate at the point of Pitot tube traverse, c.f.m.
v_n = Specific volume of air-water vapor mixture at the point of Pitot tube traverse, cu. ft. per lb. of dry air.
H_1 = Enthalpy (total heat) of air-water vapor mixture entering the air cooler, Btu. per lb. of dry air.
H_2 = Enthalpy (total heat) of air-water vapor mixture leaving the air cooler, Btu. per lb. of dry air.
t_1 = Dry bulb temperature of air entering the air cooler, °F.
t_2 = Dry bulb temperature of air leaving the air cooler, °F.

c. **Calibrated box method.**

$$q_{tha} = K_c \left(t_4 - \frac{t_1 + t_2}{2}\right) + 3.41\ E_c \qquad (32)$$

$$q_{sha} = q_{tha} \left(\frac{.24\ (t_1 - t_2)}{H_1 - H_2}\right) \qquad (33)$$

$$q_{lha} = q_{tha} - q_{sha} \qquad (34)$$

where:

q_{tha} = Net total cooling capacity, Btu. per hr.
q_{sha} = Net sensible cooling capacity, Btu. per hr.
q_{lha} = Net latent cooling capacity, Btu. per hr.
K_c = Heat leakage coefficient of calibrated box, Btu. per hr. per °F.
t_1 = Dry bulb temperature of air entering the air cooler, °F.
t_2 = Dry bulb temperature of air leaving the air cooler, °F.
t_4 = Average dry bulb temperature of air surrounding calibrated box, °F.
H_1 = Enthalpy (total heat) of air-water vapor mixture entering the air cooler, Btu. per lb. of dry air.
H_2 = Enthalpy (total heat) of air-water vapor mixture leaving the air cooler, Btu. per lb. of dry air.
E_c = Total electrical input to heaters and auxiliary equipment of calibrated box, watt-hours.

Examples

9.61 Example for a forced circulation pressure type air cooler.

A forced circulation pressure type air cooler is tested under Group III conditions by the mixing box with multiple nozzle method. Two 10 in. diameter nozzles constructed in accordance with paragraph 4.32 are used, and the static head across the nozzles is 0.82 in. of water. The air temperatures entering the air cooler are 15° dry bulb and 14.2° wet bulb, the air temperatures leaving the mixing box and nozzles are 11° dry bulb and 10.7° wet bulb. The air temperature surrounding the mixing box is 70° and the heat leakage coefficient for the mixing box has been determined in accordance with paragraph 9.41 to be 9.8 Btu. per hr. per °F. The brake horsepower input to the fan is 1.15. The air cooler is refrigerated with Freon-12 with the evaporating temperature held at 0°. The entering liquid refrigerant temperature is 92° and the leaving vapor has 4° superheat. The refrigerant flow has been determined in accordance with paragraph 4.50 to be 503 lb. per hr. The barometric pressure is 29.5 in. of mercury.

a. Refrigerant side cooling capacity calculations.
From equation (2)

$q_{tgs} = 503\ (78.75 - 29.19) = 24{,}929$ Btu. per hr.

b. Air flow calculations.
From equation (4)

$V = 1096\ \sqrt{0.82 \times 11.88} = 3410$ ft. per min.

From equation (5)

$Q_n = 0.99 \times 2 \times .545 \times 3410 = 3680$ c.f.m.

Corrected to standard air from equation (6)

$$Q_s = \frac{3680}{11.88 \times 0.075} = 4130 \text{ c.f.m.}$$

c. Air side cooling capacity calculations.
From equation (16)

$$q_{tha} = \frac{60 \times 3680}{11.88}\ (5.13 - 4.02) + 9.8\ (70-11) + 2546 \times 1.15$$

$q_{tga} = 20{,}630 + 578 + 2930 = 24{,}138$ Btu. per hr.

From equation (17)

$$q_{sga} = \frac{60 \times 3680}{11.88} \times .24\ (15-11) + 9.8\ (70-11) + 2546 \times 1.15$$

$q_{sga} = 17{,}842 + 578 + 2930 = 21{,}350$ Btu. per hr.

From equation (18)

$q_{lga} = 24{,}138 - 21{,}350 = 2788$ Btu. per hr.

$$\text{Sensible heat ratio} = \frac{q_{sga}}{q_{tga}} = \frac{21{,}350}{24{,}138} = 0.8845$$

d. Rated cooling capacity calculations.

$$\text{The rated total cooling capacity} = \frac{q_{tgs} + q_{tga}}{2}$$

$$\text{or } q_{tgr} = \frac{24{,}929 + 24{,}138}{2} = 24{,}534 \text{ Btu. per hr.}$$

The rated sensible cooling capacity = q_{tgr} × sensible heat ratio.
or $q_{sgr} = 24{,}534 \times 0.8845 = 21{,}700$ Btu. per hr.

The rated latent cooling capacity = $q_{tgr} - q_{sgr}$
or $q_{lgr} = 24{,}534 - 21{,}700 = 2834$ Btu. per hr.

9.62 Example for a forced circulation free delivery air cooler.

A forced circulation free delivery air cooler is tested under Group I conditions by the calibrated box method. The air temperatures entering the air coolers are 45° dry bulb and 42.9° wet bulb, the air temperatures leaving the air cooler are 38° dry bulb and 37.5° wet bulb. The air temperature surrounding the calibrated box is 75° and the heat leakage coefficient for the calibrated box has been determined in accordance with paragraph 9.42 to be 28.75 Btu. per hr. per °F. The measured total input to fan motor is 75 watts. The measured total electrical input to heaters is 2600 watts. The air cooler is refrigerated with 15 per cent calcium chloride brine having a specific heat of 0.79 Btu. per lb.

9.62 per °F. The entering brine temperature is 30° and the leaving brine temperature is 34°. The refrigerant flow has been determined in accordance with paragraph 4.60 to be 3300 lb. per hr. The barometric pressure is 29.0 in. of mercury.

a. Refrigerant side cooling capacity calculations:
The gross total cooling capacity is determined from equation (1)

$q_{tgs} = 3300 \times .79\ (34\text{-}30) = 10{,}428$ Btu. per hr.

The net total cooling capacity is determined from equation (3)

$q_{ths} = 10{,}428 - 75 \times 3.42 = 10{,}172$ Btu. per hr.

b. Air side cooling capacity calculations.
From equation (32)

$$q_{tha} = 28.75 \left(75 - \frac{45 + 38}{2}\right) + 3.42 \times 2600$$

$q_{tha} = 9855$ Btu. per hr.

From equation (33)

$$q_{sha} = 9855 \left(\frac{.24\ (45 - 38)}{16.50 - 14.05}\right)$$

$q_{sha} = 6758$ Btu. per hr.

From equation (34)

$q_{lha} = 9855 - 6758 = 3097$ Btu. per hr.

$$\text{Sensible heat ratio} = \frac{q_{sha}}{q_{tha}} = \frac{6758}{9855} = 0.686$$

c. Air flow calculations.
From equation (10)

$$Q_1 = \frac{9855 \times 12.82}{60\ (16.50 - 14.05)} = 860 \text{ c.f.m.}$$

Corrected to standard air from equation (11)

$$Q_s = \frac{860}{12.82 \times 0.075} = 890 \text{ c.f.m.}$$

d. Rated cooling capacity calculations.

$$\text{The rated total cooling capacity} = \frac{q_{ths} + q_{tha}}{2}$$

$$\text{or } q_{thr} = \frac{10{,}172 + 9855}{2} = 10{,}014 \text{ Btu. per hr.}$$

The rated sensible cooling capacity = q_{thr} × sensible heat ratio
or $q_{shr} = 10{,}014 \times 0.686 = 6870$ Btu. per hr.
The rated latent cooling capacity = $q_{thr} - q_{shr}$
or $q_{lhr} = 10{,}014 - 6890 = 3144$ Btu. per hr.

9.63 Example for a natural convection air cooler.
A natural convection air cooler is tested under Group I conditions by the calibrated box method. The average dry bulb temperature within the calibrated box is 45°. The air temperature surrounding the calibrated box is 78.5° and the heat leakage coefficient for the calibrated box has been determined in accordance with paragraph 9.42 to be 98 Btu. per hr. per °F. The measured total electrical input to heaters is 103 watts. The air cooler is refrigerated with Freon-12 with the evaporating temperature held at 30°. The entering liquid refrigerant is 91° and the leaving vapor has 5° superheat. The refrigerant flow has been determined in accordance with paragraph 4.50 to be 71 lb. per hr. The barometric pressure is 30.5 in. of mercury.

a. Refrigerant side cooling capacity calculations.
From equation (2)

$q_{tgs} = 71\ (82.46 - 28.95) = 3799$ Btu. per hr.

b. Air side cooling ca
From equation (22)

$q_{tga} = 98\ (78.5 - 45) + 3.41 \times 103$
$q_{tga} = 3283 + 352 = 3635$ Btu. per hr.

c. Rated cooling capacity calculations.

$$\text{The rated total cooling capacity} = \frac{q_{tgs} + q_{tga}}{2}$$

$$\text{or } q_{tgr} = \frac{3799 + 3635}{2} = 3717 \text{ Btu. per hr.}$$

Section 10. DEFINITIONS

10.01 *A forced circulation air cooler* is a factory made encased assembly including means for forcing air circulation and elements by which heat is transferred from air to refrigerant. The purpose of this unit is to transfer heat from a refrigerated space through the medium of air to the refrigerant.

10.02 *A natural convection air cooler* is an assembly as above, but depending upon natural convection fo air circulation.

10.03 *Quality of air* is the known psychrometric property of air as published in the *Refrigerating Data Book* or the U. S. Weather Bureau's publication W.B. No. 235 entitled "Psychrometric Tables."

10.04 *Dry type air cooler* is one as described in 10.01 above wherein heat transfer is *not* implemented by liquid spray during the operating period.

10.05 *Spray type air cooler* is the same as defined in 10.01 above, with the exception that the coil surfac capacity is augmented by a liquid spray during period of operation.

10.06 *Internal static pressure* for this code purpose i defined as the pressure difference due to air flo through the unit and expressed in inches of wate column.

10.07 *External static pressure* is the pressure differenc due to air flow through the distribution or duc system beyond the air cooler.

10.08 *Accuracy of readings.* Where percentage limits o readings are herein given, the reference base is th magnitude of the greater absolute reading, and no scale of instrument.

10.09 *Liquid refrigerant quantity meter* is an apparatu for determining the refrigerating capacity of a uni by measuring the circulation of the refrigerant. I includes hand-operated measuring cylinders or a integrating mechanism and supplementary device for determining the quantity of liquid refrigeran circulated in a given period of time. The refrig erating effect produced by the unit under test is th product of the lb. of refrigerant circulated per h and the difference between the heat content per l of refrigerant entering and refrigerant leaving th unit. See A.S.R.E. Circular No. 14, Sec. 24.

10.10 *A liquid refrigerant flow meter* includes a mechan ism and supplementary device for determining th instantaneous rate of circulation of liquid refrig erant. The refrigerating effect produced by the a cooler under test is the product of the lb. of refrig erant circulated per hr. and the difference betwee the heat content per lb. of refrigerant entering an refrigerant leaving the air cooler. See A.S.R. Circular No. 14, Sec. 25.

10.11 *A free delivery type air cooler* takes air from ar discharges it directly to the space to be treate without any element external to the cooler imposin air resistance.

10.12 *A pressure type air cooler* is for use with one more external elements which impose air resistanc

10.13 *The total cooling effect of an air cooler* is the diffe ence between the total heat content of the dry a and water vapor mixture entering the cooler p

0.13 hour and the total heat content of the dry air and water vapor mixture leaving the cooler per hour, expressed in Btu. per hr.

0.14 *The sensible cooling effect* of an air cooler in Btu. per hr. is the difference between the total cooling effect and the dehumidifying effect.

0.15 *The sensible heat ratio* of an air cooler is the ratio of the sensible cooling effect to the total cooling effect.

0.16 *Evaporative equilibrium* of a wet bulb instrument is the condition attained when the wetted wick has reached a stable and constant temperature. When the instrument is exposed to air at velocities over 50 ft. per min. this temperature may be considered the true wet bulb temperature.

0.17 *A volatile refrigerant* is one which changes from the liquid to the vapor phase in the process of absorbing heat within the air cooler.

0.18 *A non-volatile refrigerant* is one which remains a liquid during the process of absorbing heat within the air cooler.

0.19 *Forced air circulation* is that caused by a difference of pressure produced by a fan or blower.

0.20 *Natural convection circulation* of air is that produced by gravity in response to a temperature difference of the air itself.

ppendix A. Instructions for Using the Wet Bulb Psychrometer at Temperatures Below Freezing*

election of Thermometers

At low temperatures, the amount of moisture in the tmosphere is greatly reduced and wet bulb depressions re correspondingly small. Thus, while thermometers raduated in degrees or half-degrees are satisfactory at omfort temperatures, they are inadequate in the frost one. Table A1 shows the effect of thermometer accuracy n the relative humidity readings at various dry bulb emperatures. It will be noted that a thermometer error f 0.2° causes a discrepancy of only 1 per cent at 60°F. ry bulb temperature but this increases to 5.2 per cent at °F. For the accuracy usually required in engineering ractice, the wet bulb and dry bulb thermometers should e graduated to 0.2° for temperatures near 20°F. and to 1° or finer for temperatures near 0°F. Thermometers hould be checked against a precision standard or against ach other to determine any difference in their calibration.

Table A1.

Dry bulb emp., °F.	*Deviation of the percent relative humidity corresponding to various thermometer errors* (Δt)			
	$\Delta t = 1°$	$\Delta t = 0.5°$	$\Delta t = 0.2°$	$\Delta t = 0.1°$
60	5.0%	2.5%	1.0%	0.5%
40	8.0	4.0	1.6	0.8
20		6.0	2.4	0.2
0			5.2	3.6
—10				4.3

he Ice Coated Bulb

The wick covering for the wet bulb thermometer at mperatures below freezing no longer serves its usual rpose, for ice, unlike water, does not respond to capilry forces. True wet bulb readings are obtained only hen the surface of the wick is completely covered with layer of ice. The ice held within the wick is useless.

As a result of the reduced vapor pressure at low temratures, a longer time is necessary to reach equilibrium an at higher temperatures. This condition is offset, wever, by the ice remaining on the bulb for a much longer period of time. Readings of the wet bulb thermometer must be continued over a sufficiently long period to insure that equilibrium has been reached.

Careful tests indicate that most reliable results are obtained by discarding the wick and freezing a layer of ice directly on the thermometer bulb. The ice film is best formed by dipping the chilled thermometer into distilled water at approximately 32°F. The thermometer is then removed from the water and the film allowed to freeze. The process may be repeated several times if necessary to build up a suitable film thickness.

In order to prevent conduction of heat along the glass stem of the thermometer to the bulb, it is important that the ice film cover about one inch of the stem as well as the bulb.

Super-Cooled Water

The tendency of water to resist freezing at temperatures considerably below freezing is well known. This condition of super-cooled water can and does exist on the wick of the wet bulb thermometer and may cause large errors. It should be borne in mind that psychrometric observations made with super-cooled water are entirely satisfactory if the proper physical constants are used in calculating the results or if a psychrometric chart is available showing wet bulb temperatures with super-cooled water. Such a chart is presented in Fig. A1.

The usual procedure of freezing water on the wick has the disadvantage of first going through a period of subcooling before finally freezing. In this process, the temperature falls slowly to some temperature below 32°F., then when freezing starts, the temperature rises quickly to 32° and remains there during the freezing process. Finally, the temperature slowly drops to a state of equilibrium. This final temperature may be either higher or lower than the previous low point.

The following procedure is recommended: At temperatures well below freezing, discard the wick and coat the bulb of the thermometer with ice. If necessary, freezing can be initiated by touching the bulb to a piece of ice or frost. At temperatures only slightly below freezing where difficulty may be experienced in obtaining an ice coating, the wick should be wetted with super-cooled water and calculations made accordingly.

Notes on the Wet Bulb Wicks

A suitable material for the wick is cotton tubing of a fairly soft, fine mesh weave. Before use, the wick should be thoroughly cleaned by washing or boiling. A snug fit on the thermometer bulb is necessary and in order to prevent conduction of heat from the stem, the wick should cover about 1 in. of the stem as well as the bulb. With continued use, wicks become encrusted with impurities that interfere with the proper action. It is, therefore, highly important that wicks be frequently cleaned or replaced.

Radiation from Coils and Other Objects

If the wet bulb is located close to coils or other surfaces (the temperature of which is different from the dry bulb temperature), a radiation shield should be interposed. The outside surface of this shield should be polished metal having a low coefficient of emissivity. The inside surface facing the wet bulb should be painted or otherwise reduced to black body conditions.

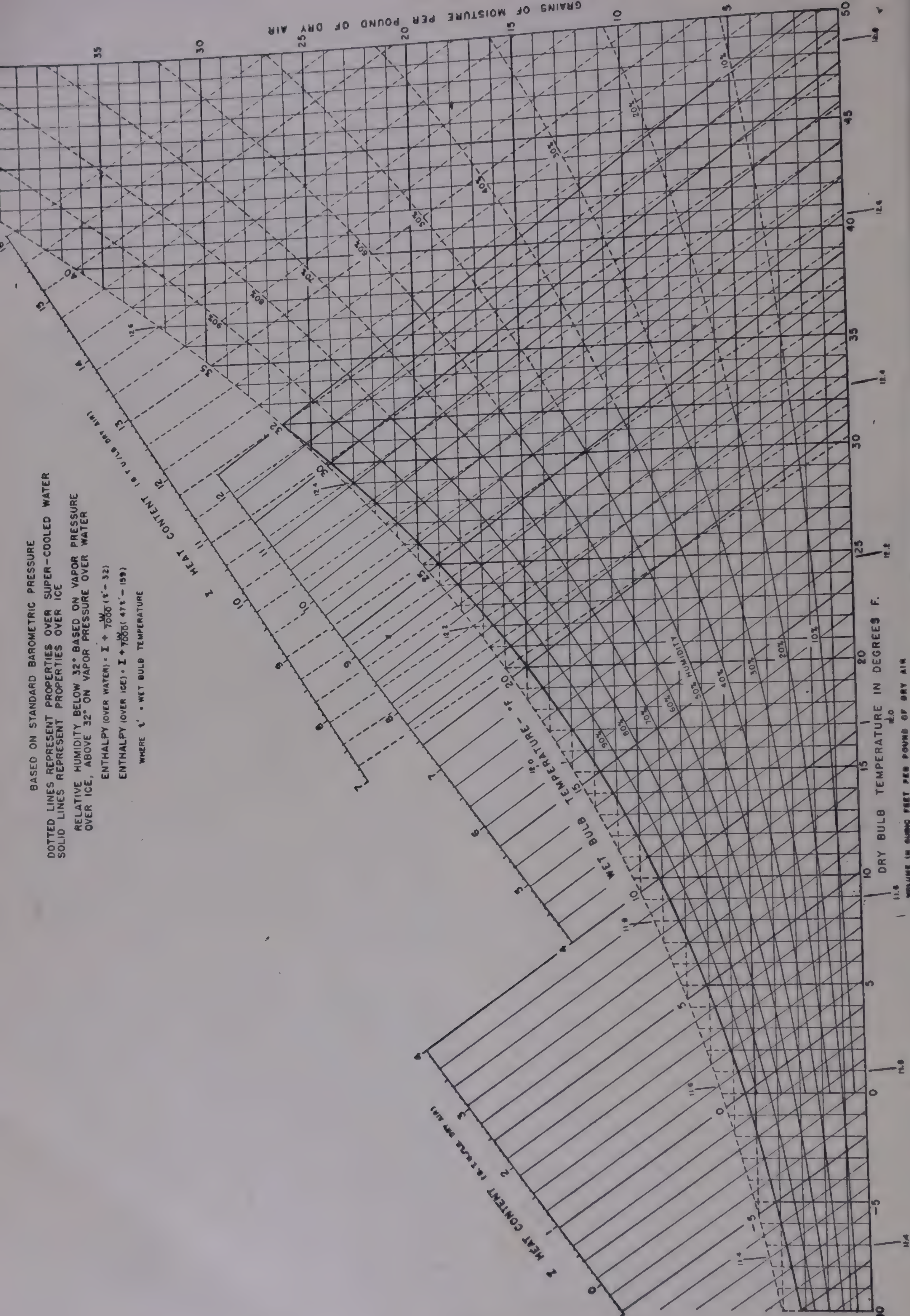
Wet bulbs located in air ducts need not be shielded from the duct wall if the wall is at or near dry bulb temperature. If necessary, the outside of the duct should be insulated to avoid large temperature gradients between the duct wall and the inside dry bulk temperature.

The Effect of Air Stream Velocity

The wet bulb thermometer receives radiant heat from the surrounding objects at dry bulb temperature. The amount of heat received depends upon the difference between the wet and dry bulb temperatures and also upon the absolute temperature. The rate of dissipation of this

---

Condensed from the paper "Psychrometry in the Frost Zone," D. D. le, *Refrig. Eng.*, vol. 48, pp. 291-300. 1944.


BASED ON STANDARD BAROMETRIC PRESSURE
DOTTED LINES REPRESENT PROPERTIES OVER SUPER-COOLED WATER
SOLID LINES REPRESENT PROPERTIES OVER ICE
RELATIVE HUMIDITY BELOW 32° BASED ON VAPOR PRESSURE OVER ICE, ABOVE 32° ON VAPOR PRESSURE OVER WATER
ENTHALPY (OVER WATER) = I + W/7000 (t' − 32)
ENTHALPY (OVER ICE) = I + W/7000 (.47t' − 159)
WHERE t' = WET BULB TEMPERATURE
HEAT CONTENT (B.T.U./LB DRY AIR)
WET BULB TEMPERATURE — °F
DRY BULB TEMPERATURE IN DEGREES F.
VOLUME IN CUBIC FEET PER POUND OF DRY AIR
GRAINS OF MOISTURE PER POUND OF DRY AIR
HUMIDITY
-10
-5
0
5
10
15
20
25
30
35
40
45
50
11.4
11.6
11.8
12.0
12.2
12.4
12.6
90%
80%
70%
60%
50%
40%
30%
20%
10%

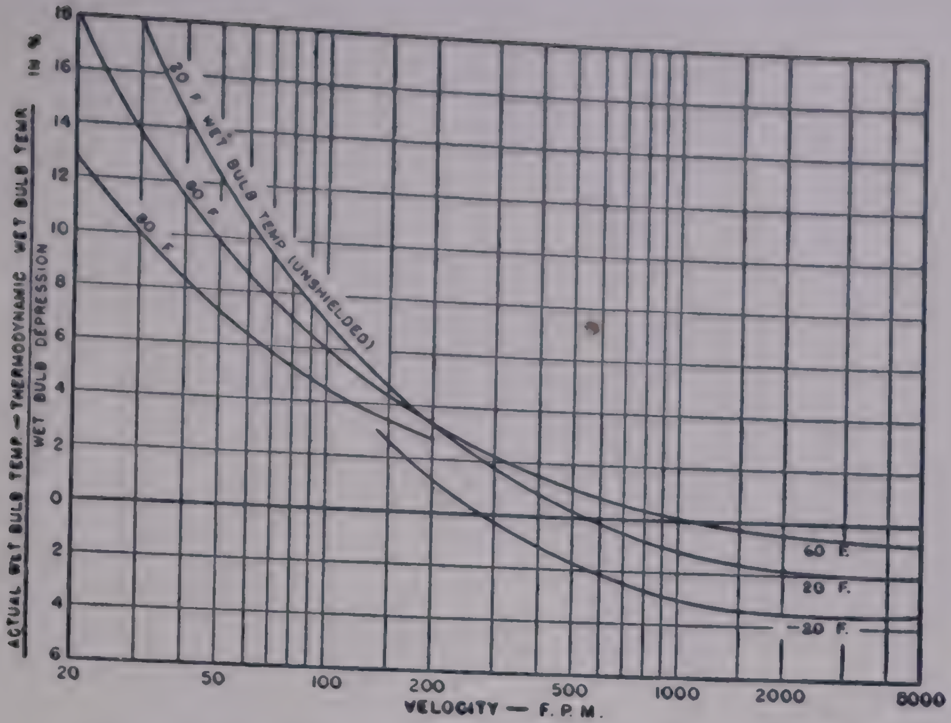


FIG. A2. THE EFFECT OF AIR STREAM VELOCITY ON THE WET BULB PSYCHROMETER. (FOR CONVENTIONAL SIZE THERMOMETER BULB.)

eat depends largely upon the air velocity. These factors re inter-related with each other and with certain other ictors in a rather complex manner. In Fig. A2, the error rom this cause is shown as a per cent of the wet bulb epression. It will be noted that, except for very low mperatures, the error can be considered negligible for ost engineering purposes if the air velocity is maintained bove 300 ft. per min.

Use of Thermocouples

Thermocouples are entirely satisfactory for taking psyrometric readings if suitable equipment is available and oper precautions are taken in their use. The wet and y bulb thermocouples may be connected in series to give direct reading of the depression and, when measured a potentiometer of suitable accuracy, such a series uple forms one of the most reliable methods of obtaing psychrometric data.

Thermocouple wires most commonly used in refrigerion practice are copper and constantan or iron and conntan. Copper-constantan is preferred for freem from rust. Copper has a high thermal conductivity d may introduce errors due to the conduction of heat om the surrounding air at dry bulb temperature to the wer wet bulb temperature. This condition is best remied by selecting the copper wire about twelve sizes smaller than the constantan. For example, No. 36 copper wire should be selected for use with No. 24 constantan.

The wick must fit snugly over the thermocouple wire. Small size cotton tubing can be obtained for this purpose. For No. 24 constantan and No. 36 copper wire, the wick should cover about 1 inch of the lead wires beyond the junction, care being taken to insulate the leads from each other to prevent short circuiting. With larger size wire, a much longer length of lead must be included within the wick. At temperatures well below freezing, the wick may be discarded and ice frozen directly on the thermocouples.

Any number of thermocouples may be connected in series to form a thermopile in which half of the junctions are exposed to the dry bulb temperature and the other half to the wet bulb temperature. This arrangement greatly multiplies the accuracy of the thermocouple measurement and forms one of the most precise methods of determining the wet bulb depression.

The small size of the thermocouple makes it particularly suited for psychrometric determinations at low air stream velocities. The velocity effect, as shown in Fig. A2, is calculated for a wet bulb diameter of ¼ in. For smaller diameters, the velocity requirement is reduced proportionately. Thus, a wet bulb thermocouple having a diameter of 1/16 in. will have the same correction at 100 ft. per min. as the ¼-in. thermometer has at 400 ft. per min.

ASA
Reg. U. S. Pat. Off.
ASRE Standard 26-R
ASA B59.1-1950
UDC 621.56:629.12

American Standard

RECOMMENDED PRACTICE FOR MECHANICAL REFRIGERATION INSTALLATIONS ON SHIPBOARD

SPONSOR—American Society of Refrigerating Engineers

Approved by the American Society of Refrigerating Engineers
December 8, 1948

Approved by American Standards Association, Incorporated, November 3, 1950

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 WEST 40TH STREET NEW YORK 18, N. Y.
Price $1.00

Joint Committee on Recommended Practice for Mechanical Refrigeration Installations on Shipboard

W. L. KELLER, *Chairman*
United Fruit Company

W. F. R. KARSTEN, *Vice Chairman*
General Electric Company

Cecil Boling, Cecil Boling Company

Seymour W. Brown, Carrier Corporation

R. A. Chadburn, Heat Exchanger Company
Subcommittee Chairman, 1945-1947

J. R. Chamberlain, York Corporation

Harry D. Edwards, The Linde Air Products Company

I. M. Fisher, General Electric Company

A. R. Gatewood, American Bureau of Shipping
Subcommittee Chairman, 1949, 1950

C. W. Hudzietz, Henry Valve Company

C. Macpherson, Lloyds Register of Shipping

H. E. Parker, Bethlehem Steel Co., Central Technical Dept., Shipbuilding Div.

R. C. Robertson, Superior Valve and Fittings Company

Arthur C. Rohn, U. S. Dept. of Commerce, Federal Maritime Administration

R. A. Smyth, Captain, U. S. Coast Guard

R. J. Thompson, Kinetic Chemicals, Inc.

August Ulbert*, Alco Valve Company
Subcommittee Chairman, 1947, 1948, 1949

L. L. Westling, Matson Navigation Company

ASRE Standards Committee, 1948

R. C. Jordan, *Chairman*

D. D. Wile, *Vice Chairman*

R. E. Backstrom

J. R. Chamberlain

H. M. Hendrickson

G. B. Priester

M. F. Tokach

---

* Deceased

ASA Standard B-59

Recommended Practice for Mechanical Refrigeration Installations on Shipboard

CONTENTS

1. Scope and Purpose
2. Definitions
3. Refrigerant Classification
4. Use of Refrigerants
5. High Side and Pressure Vessels
6. Piping, Valves, Fittings, and Controls
7. Evaporators and Cooling Coils
8. Installation and Tests
9. Spare Parts and Tools
10. Operation, Maintenance, and Inspection

1. SCOPE AND PURPOSE

1.1 These Recommendations have been drawn up to serve as a guide for the design, construction, installation, operation, tests, and inspection of *mechanical refrigeration* equipment for the extraction of heat aboard *merchant vessels*. The systems included among others are those for the carrying of *refrigerated cargo*, the preparation and preservation of food, and the cooling or dehumidification of air for circulation to or within ships *compartments* for whatever purpose.

1.2 Recommendations are limited to systems obtaining cooling effect by the vaporization of a refrigerant, the compression of same by power driven mechanical equipment and condensing of the compressed gas for re-use.

1.3 Recommendations cover machinery, prime movers, *pressure vessels*, heat exchange apparatus, *piping*, valves, fittings, etc., which during operation of the system come into contact with the *primary refrigerant* or with any circulated secondary fluid (*brine*, water, etc.) used for transfer of heat and any accessories, controls, and apparatus affecting the performance or safe operation thereof.

1.4 Recommendations indicate what is considered good engineering practice pertaining to safety of the personnel and of the ship itself, as well as reliability, durability, and adequacy of the apparatus to maintain the *refrigerated cargo* and refrigerated *ships stores* in a satisfactory condition for the extent of the voyage and the loading and unloading periods. It is not intended, however, to cover all details of design and construction, nor to limit development and adoption of new methods, designs, or forms of construction which can be demonstrated to conform to the intent of the Recommendations as stated above.

1.5 Recommendations are intended to supplement and amplify regulations and requirements set forth by the U. S. Coast Guard, the Classification Societies, and other marine regulatory bodies with particular reference to installation of *mechanical refrigeration* equipment and the insulation of refrigerated spaces aboard ship. The above mentioned regulations have been given full consideration in the preparation of these Recommendations.

2. DEFINITIONS

2.1 Definitions given below are compiled for the purpose of clarifying these Recommendations only, and are not necessarily to be considered as authoritative definitions for general use throughout the field. Terms defined are printed in italics wherever they appear in the text.

2.2 ***Air Conditioning.*** The simultaneous control of temperature, humidity, purity, and motion of the air within a *compartment.*

2.3 ***Ambient Air Temperature*** (a) When used in connection with machinery, temperature of air in the *engine room* or *refrigeration machinery compartment* surrounding such machinery.
(b) Temperature of the surrounding air outside the ship or in adjacent spaces (when used in connection with refrigerated *compartments).*

2.4 ***Approved.*** Having been considered and accepted by the authorities having jurisdiction. (May include the Classification Society, U. S. Coast Guard, U. S. Dept. of Commerce Federal Maritime Administration, shipbuilder, naval architect, owner, operator, or others.)

2.5 ***Auxiliary Machinery Space*** A *compartment* assigned to machinery, generally other than main propulsion equipment.

2.6 ***Back Pressure.*** *Operating pressure* measured in the suction line at the inlet to the *compressor.*

2.7 ***Brazed Joint (High Temperature).*** A gas tight joint obtained by the joining of metal parts with alloys which melt at temperatures higher than 1500 F but less than the melting temperatures of the joined parts.

2.8 ***Brazed Joint (Low Temperature).*** A gas tight joint obtained by the joining of metal parts with metallic mixtures or alloys which melt at temperatures below 1500 F but above 1000 F.

2.9 ***Breather Plug.*** A removable plug, cap, or other means of venting a space containing insulating material through vapor tight sheathing to the interior of refrigerated *compartment.*

2.10 ***Brine.*** Any liquid cooled by the refrigerant and used for the transmission of heat without a change in its state, having no flash point or a flash point above 150 F determined by American Society for Testing Materials Method D93.

2.11 ***Brine Cooler.*** Heat exchange equipment used for cooling *brine* by the evaporation of refrigerant. (See also *Evaporator*).

2.12 ***Brine Expansion Tank.*** A vented reservoir in a closed circulating *brine* system for the accommodation of volume expansion of *brine* due to temperature change.

2.13 ***Brine Return Tank.*** A reservoir in an open circulating *brine* system for the storage of *brine* at the pump suction and for inspection of the condition and flow of *brine*.

2.14 ***By-pass.*** A pipe or *duct*, usually controlled by valve or *damper*, for conveying a fluid around an element of a system.

2.15 ***Capacity of Refrigerating System.*** The cooling effect produced by the change in enthalpy (formerly called heat content) between the refrigerant liquid entering the expansion valve and the vapor leaving the *evaporator*, generally measured in Btu per hour or tons of refrigeration.

2.16 ***Cargo Battens.*** Protection members applied permanently to the interior of a refrigerated *compartment* to provide air space between the stowed cargo and the sides of the *compartment*.

2.17 ***Centrifugal Compressor.*** A nonpositive displacement *compressor* which depends for pressure rise, at least in part, on centrifugal force.

2.18 ***Charging Valve.*** A valve used to charge or add refrigerant to the system, or add oil to the *compressor* crankcase.

2.19 ***Check Valve.*** A valve used to prevent reverse flow of a fluid or gas.

2.20 ***Chilled Cargo.*** Cargo maintained at an assigned temperature above its freezing point.

2.21 ***Closed Brine System.*** *Refrigerating system* in which the circulating *brine* is completely enclosed and shut off from the atmosphere throughout the cycle except for a vented expansion tank at the high point of the system.

2.22 ***Compartment.*** An enclosed subdivision within the ship (corresponding to a room).

2.23 ***Compressor.*** A specific machine, with or without accessories, for compressing a given refrigerant vapor.

2.24 ***Condenser.*** A vessel or arrangement of pipe or tubing in which compressed refrigerant vapor is liquefied by the removal of heat.

2.25 ***Condensing Unit.*** An assembly consisting of *compressor*, *condenser*, receiver, prime mover drive, and regularly furnished accessories.

2.26 ***Container.*** A cylinder for the transportation of refrigerant.

2.27 ***Cooling Coil.*** An arrangement of pipe or tubing which transfers heat from the material or space cooled to the *primary refrigerant* or *brine*.

2.28 ***Cooling Water.*** Water circulated for the purpose of removing the heat of compression and condensing the refrigerant.

2.29 ***Damper.*** A device to regulate or stop the flow of air in a *duct*.

2.30 ***Deck Grating.*** An open structure installed to provide an air space between the stowed cargo and the deck.

2.31 ***Defrosting.*** Removing of accumulated ice or frost from the surface of a *cooling coil*.

2.32 ***Dehydrator.*** A device provided with chemicals for removing moisture from the refrigerant.

2.33 ***Design Pressure.*** The theoretical bursting pressure of the weakest part of the *pressure vessel* or other pressure part under consideration, divided by its factor of safety.

2.34 ***Dial Thermometer (remote reading).*** An instrument for indicating temperature on a dial by a gage movement actuated by the expansion or contraction of mercury or other fluid through a capillary system from a remotely located temperature sensitive bulb.

2.35 ***Direct Connected.*** Driver and driven equipment with positive connection for rotation at the same speed.

2.36 ***Direct Expansion System.*** *Refrigerating system* in which the *evaporator* is in direct contact with the refrigerated material or space or is located in air circulating passages communicating with such spaces.

2.37 ***Duct, (air).*** A tube or conduit used for conveying air.

2.38 ***Dunnage.*** Strips of wood used in stowing cargo to provide air space between pieces or packages.

2.39 ***Electric Refrigerator.*** A completely self-contained unit comprising an insulated cabinet, *evaporator*, and electric motor driven *condensing unit*.

2.40 ***Electric Water Cooler.*** A completely self-contained unit comprising *evaporator*, water connections, and electric motor driven *condensing unit* for cooling drinking water.

2.41 ***Electrolytic Brine.*** Any *brine* capable of causing decomposition of one of two dissimilar metals by electrolysis.

2.42 ***Engine Room.*** A *compartment* containing the ship's propulsion machinery and auxiliaries.

2.43 ***Evaporator.*** That part of the system in which the refrigerant is vaporized to produce refrigeration.

2.44 ***Evaporator Pressure Regulator.*** A control valve in the suction line for automatically maintaining a predetermined minimum pressure in the *evaporator*.

2.45 ***Expansion Coil.*** An *evaporator* constructed of pipe or tubing.

2.46 ***Filter.*** A device for separating dirt and other solid matter from refrigerant, oil, or air.

2.47 ***Flash Point.*** Temperature of a flammable material at which there is sufficient vaporization to permit igniting the vapor, but not sufficient vaporization to support combustion of the material, as determined by A.S.T.M. Method D93.

2.48 ***Float Valve.*** A regulating valve connected to a buoyant ball actuated by liquid level.

2.49 ***Frozen Cargo.*** Cargo maintained at an assigned temperature below its freezing point.

.50 ***Gage Glass.*** A device for direct observation of the level of a confined liquid.

.51 ***Hand Expansion Valve.*** A manually operated needle type valve for controlling the flow of liquid refrigerant to an *evaporator*.

.52 ***Head Pressure.*** *Operating pressure* measured in the discharge line at the outlet from the *compressor*.

.53 ***Heat Interchanger.*** A device to transfer heat from the liquid refrigerant to the suction gas.

2.54 ***High Side.*** All parts of the system under condenser pressure or higher.

2.55 ***Hot Gas Line.*** A line used to convey discharge gas from the *compressor* to the *evaporator* for the purpose of *defrosting*.

2.56 ***Hydrocarbon Refrigerant.*** Refrigerant containing only hydrogen and carbon, such as ethane, propane, etc.

2.57 ***Indirect System.*** *Refrigerating system* in which the *evaporator* is utilized to cool *brine* or air which in turn is used for removing heat from the material or space refrigerated.

2.58 ***Flammable Refrigerant.*** Any refrigerant shall be classified as flammable if (a) a mixture of 13 per cent or less (by volume) with air forms a flammable mixture or (b) the flammability range with air is greater than 12 per cent regardless of the lower limit.

2.59 ***King Valve.*** A *stop valve* in the main liquid line leaving the *liquid receiver*.

2.60 ***Leak Detector.*** A means for determining the location of refrigerant leakage.

2.61 ***Liquid Receiver.*** A vessel permanently connected to a system by inlet and outlet pipes for storage of a liquid refrigerant.

2.62 ***Low Side.*** All parts of the system under *evaporator* pressure.

2.63 ***Manufacturer.*** For the purpose of this Code, a company or organization which evidences its responsibility by affixing its name or nationally registered trade mark or trade name to the refrigeration equipment concerned.

2.64 ***Maximum Working Pressure.*** See *Design Pressure.*

2.65 ***Mechanical Joint.*** A gas tight joint obtained by the joining of metal parts through a positive holding mechanical construction (such as flanged joint, screwed joint, flared joint).

2.66 ***Mechanical Refrigeration.*** Process of obtaining a cooling effect by the vaporization of a refrigerant at a temperature below that of the material or space being cooled, the compression of the resulting vapor by power driven mechanical equipment, and condensing of the compressed gas for re-use.

2.67 ***Merchant Vessel.*** A ship which carries cargo or passengers on a commercial basis.

2.68 ***Noncondensable Gas.*** A foreign gas (such as air or nitrogen) in the refrigerant system which will not condense at the existing condensing temperature and hence will increase the *head pressure* above that corresponding to the condensing temperature.

2.69 ***Oil Charging Pump.*** A hand operated pump for forcing oil into the *compressor* crank case when the crank case pressure is above atmospheric.

2.70 ***Oil Separator.*** A device for separation of oil from refrigerant vapor.

2.71 ***Oil Still.*** A device to separate oil from refrigerant by a distillation process.

2.72 ***Open Brine System.*** *Refrigerating system* in which the circulating *brine* returns to an open tank which serves as a reservoir for the pump suction and as an inspection tank for the condition and flow of *brine*.

2.73 ***Operating Pressure.*** The pressure that would be indicated by a gage in the part of the system involved when the system is in normal operation.

2.74 ***Piping.*** Pipe or tube for interconnecting the parts of a *refrigerating system*. (See also 2.78.)

2.75 ***Pressure Limiting Device.*** A pressure responsive mechanism designed to stop the operation of the *compressor* automatically at a predetermined pressure.

2.76 ***Pressure Relief Device.*** A valve or rupture member designed to relieve excessive pressure automatically.

2.77 ***Pressure Relief Valve.*** A valve held closed by a spring or other means and designed to relieve pressure automatically in excess of its setting, and to reseat again when pressure has been reduced to a safe point.

2.78 ***Pressure Vessel.*** Any part of the system containing refrigerant except *compressor, expansion coils*, or *piping*. (Any container six inches or less in inside diameter should be considered part of the *piping* and the part of these Recommendations covering *piping* should apply thereto. See also 2.74.)

2.79 ***Primary Refrigerant.*** A substance (or medium) used to produce refrigeration by absorbing heat in its expansion or vaporization as distinguished from *brine*.

2.80 ***Purging.*** Removing of *noncondensable* gases from the system.

2.81 ***Purging Device.*** An automatic or semiautomatic device which collects *noncondensable gas* from the *condenser* or receiver, condenses some of the condensable refrigerant therefrom, and relieves the remainder to the atmosphere. (May also be a hand operated valve.)

2.82 ***Purge Valve.*** A manually operated valve located at a high point of the refrigerant system to draw off *noncondensable gases.*

2.83 ***Rat-Proof.*** Acceptable under the rat-proofing rules of U. S. Public Health Service.

2.84 ***Refrigerated Cargo.*** Cargo maintained at an assigned temperature by means of *mechanical refrigeration.*

2.85 ***Refrigerating Machinery Space.*** A space assigned to *high side* refrigeration equipment and *brine* cooling, heating, and circulating equipment when used.

2.86 ***Refrigerating System.*** All equipment installed for the purpose of circulating *primary refrigerant* or *brine* for extraction of heat from a material or space.

2.87 ***Resistance Thermometer.*** An instrument for measuring temperatures at remote points as a function of the electrical resistance of an element placed at the point where the temperature is to be determined.

2.88 ***Rupture Disc.*** A safety device which will rupture at a predetermined pressure.

2.89 ***Safety Head.*** In a *compressor*, a cylinder head held in place by a spring of such strength that it will not be compressed during normal operation, but will be compressed by solid matter or liquid coming between it and the piston, thereby protecting the *compressor*.

2.90 ***Scuttlebutt.*** An *evaporator* or cooler used for cooling drinking water at a central point.

2.91 ***Ships Stores.*** Material stowed aboard a ship for use or consumption by crew or passengers.

2.92 ***Sight Glass.*** A device for direct visual indication of the presence, level, flow, or condition of a confined fluid.

2.93 ***Soldered Joint.*** A gas tight joint obtained by the joining of metal parts with metallic mixtures or alloys which melt at temperatures below 1000 F. See 6.7.

2.94 ***Solenoid Valve.*** A valve which is closed by gravity, pressure, or spring action and opened by the movement of a plunger due to the magnetic action of an electrically energized coil. For the purpose of these Recommendations a valve which is normally open and becomes closed when energized will be known as a reverse acting *solenoid valve.*

2.95 ***Stop Valve.*** A manually operated valve for stopping flow of fluid.

2.96 ***Strainer.*** A device for withholding foreign matter from a flowing liquid or gas.

2.97 ***Thermostat.*** A temperature sensitive device used to actuate a control mechanism.

2.98 ***Thermostatic Expansion Valve.*** An automatic valve equipped with a temperature sensitive bulb to regulate the flow of refrigerant to the *evaporator* according to changes in superheat of the refrigerant suction gas leaving the *evaporator*.

2.99 ***Water Cooler.*** Same as *Brine Cooler*. The term is applied where water is used as a secondary refrigerant. See also *Scuttlebutt, Electric Water Cooler.*

2.100 ***Water Regulator Valve.*** An automatic valve to control the flow of *cooling water* through a *condenser* to maintain a predetermined *head pressure.*

2.101 ***Welded Joint.*** A gas tight joint obtained by the joining of metal parts in the plastic or molten state.

3. REFRIGERANT CLASSIFICATION

3.1 For the purpose of these Recommendations refrigerants are divided into three groups. Group 1 contains those which have negligible toxicity and are nonflammable. Group 2 refrigerants are either toxic or flammable or both. Group 3 refrigerants are highly flammable and explosive. For further reference on toxicity and flammability see Underwriters Laboratories Reports MH-2256, MH-2375, MH-2630, MH-3072, MH-3134, also U. S. Bureau of Mines Reports RI-3013, RI-3042, and RI-3185.

3.2 ***Group 1***

Carbon Dioxide	CO_2
Dichloromethane (Methylene Chloride)	CH_2Cl_2
Freon-11 (Trichloromonofluoromethane)	CCl_3F
Freon-12 (Dichlorodifluoromethane)	CCl_2F_2
Freon-21 (Dichloromonofluoromethane)	$CHCl_2F$
Freon-22 (Monochlorodifluoromethane)	$CHClF_2$
Freon-113 (Trichlorotrifluoroethane)	$C_2Cl_3F_3$
Freon-114 (Dichlorotetrafluoroethane)	$C_2Cl_2F_4$

3.3 ***Group 2***

Ammonia	NH_3
Dichloroethylene	$C_2H_2Cl_2$
Ethyl Chloride	C_2H_5Cl
Methyl Chloride	CH_3Cl
Methyl Formate	$HCOOCH_3$
Sulphur Dioxide	SO_2

3.4 ***Group 3***

Butane	C_4H_{10}
Ethane	C_2H_6
Isobutane	$(CH_3)_3CH$
Propane	C_3H_8

4. USE OF REFRIGERANTS

4.1 It is recommended, for reasons of safety, that refrigerants for new installations be confined to Group 1, with the exception of ammonia when used with *indirect systems.* Other refrigerants listed in Groups 2 and 3 may be used for special purposes provided *approved* safety precautions have been incorporated in the design and installation of the system.

4.2 Machinery *compartments* containing equipment for ammonia should be fitted with a sprinkler system providing an effective water spray and having a remote control device located outside the *compartment* as *approved.*

4.3 Where ammonia is used in new installations as mentioned in paragraph 4.1 above, all parts containing refrigerant should be confined in a gas tight *compartment* provided with supplementary access, independent of engine room, crew's quarters, passageways, or other adjoining living or working spaces, and fitted with means for continuously operating natural supply and forced exhaust ventilation. The amount of such ventilation should be computed on the basis of 200 cubic feet per minute for each 100 pounds of refrigerant charge up to 1000 pounds, plus 80 cubic feet per minute for each 100 pounds by which the total charge exceeds 1000 pounds. The ductwork from the fan outlet should terminate at a height above deck such as to prevent entry of fumes into navigating and living quarters and any inlet openings for ventilation air. The amount of ventilation should be based on the refrigerant charge of the largest system in the machinery *compartment.* Controls for ventilation should be located outside the *refrigerating machinery space.*

4.4 Where ammonia refrigerant is used, at least two masks or helmets as *approved* by the Bureau of Mines of the U. S. Department of the Interior should be provided and stowed in a cabinet at a safe distance outside of the *refrigerating machinery space.* Canisters or cartridges should be renewed immediately after having been used or the seal broken and if unused should be renewed at least once every two years. The date of filling should be marked in the space provided.

4.5 All refrigerant stowed on board except that contained in the system should be kept in *containers* as *approved* by the U. S. Interstate Commerce Commission. Any refrigerant withdrawn from the system for whatever purpose or duration should be stowed only in such *contain-*

ers. Care should be taken not to fill *containers* beyond the rated capacity.

4.6 Not more than 300 pounds of ammonia refrigerant should be stowed in *refrigerating machinery space*. Any ammonia refrigerant carried in excess of 300 pounds should be stowed in a location remote from machinery spaces, engine and boiler rooms, navigating quarters, and living quarters.

4.7 It is recommended that materials such as warning agents, antifreeze solutions, etc. should not be added to any *primary refrigerant* system.

4.8 Carbon dioxide should not be used in an atmospheric free state unless proper precautions are taken to prevent hazard to life through a deficiency of oxygen.

4.9 Solid carbon dioxide or dry ice should not normally be used to cool *refrigerated cargo* or *ships stores* for the reason stated in paragraph 4.8 above.

5. HIGH SIDE AND PRESSURE VESSELS

Pressure Vessels

5.1 Design, manufacture, and factory testing of *pressure vessels* containing *primary refrigerants* are included in this section.

5.2 All *pressure vessel* materials should be suitable for the fluid used such as *primary refrigerant*, *brine*, water, oil, etc.

No material should be used that will unduly deteriorate due to chemical action of *primary refrigerant*, oil (or combination of both), *brine*, or *cooling water*.

No materials should be used that become unduly brittle at the lowest intended operating temperature.

Any combination of materials should be carefully selected by reference to their relative position in the electromotive series in order to minimize electrolytic action.

General Requirements for Pressure Vessels

5.3 *Pressure vessels* should be properly supported to prevent damage during normal operations aboard ship. Such protective measures should not interfere with operation.

5.4 Brass, bronze, or copper or its alloys should not be used when temperatures over 400 F may be encountered or with ammonia refrigerant.

No brazed joints should be used where ammonia is the refrigerant.

5.5 All bolts and nuts should have hexagonal heads in accordance with ASA B18.2 Heavy Series. Wherever possible through bolts should be used. In the event that through bolts cannot be used, studs and nuts should be provided.

5.6 All *pressure vessels* should conform to the requirements of the Classification Societies and U.S.C.G. as regards the inspection and testing of materials, procedure as to welding, brazing, or other manufacturing processes and to the testing of the finished vessels. It is recommended that, in order to protect the makers of refrigerating machinery and appliances, the steel works and foundries supplying materials to the makers should provide certificates of tests or affidavits for the materials supplied irrespective of whether the installation is being built under the survey of a Classification Society.

5.7 The inside of *pressure vessels* and appurtenances thereto should be smooth, clean, and free from all foreign matter when installed.

5.8 All equipment should be designed for proper operation under conditions of a momentary roll of 30° to either side and a permanent list of 15° and a permanent trim of 5° fore and aft, plus pitch of 5°.

5.9 *Pressure vessels* should be designed for not less than the *design pressure* shown in Table 5-1.

Table 5-1.

Refrigerant	*Formula*	*Design pressure, psi*	
		High side	*Low side*
GROUP 1			
Carbon Dioxide	CO_2	1500	1000
Methylene Chloride (Dichloromethane)	CH_2Cl_2	30	30
Freon-11 (Trichloromonofluoromethane)	CCl_3F	30	30
Freon-12 (Dichlorodifluoromethane)	CCl_2F_2	225	150
Freon-21 (Dichloromonofluoromethane)	$CHCl_2F$	70	40
Freon-22 (Monochlorodifluoromethane) Water Cooled *Condensers* Only	$CHClF_2$	300	150
Freon-113 (Trichlorotrifluoroethane)	$C_2Cl_3F_3$	30	50
Freon-114 (Dichlorotetrafluoroethane)	$C_2Cl_2F_4$	50	50
GROUP 2			
Ammonia	NH_3	300	150
Dichloroethylene	$C_2H_2Cl_2$	30	30
Ethyl Chloride	C_2H_5Cl	60	50
Methyl Chloride	CH_3Cl	210	125
Methyl Formate	$HCOOCH_3$	50	50
Sulphur Dioxide	SO_2	170	95
GROUP 3			
Butane	C_4H_{10}	95	50
Ethane	C_2H_6	1200	700
Isobutane	$(CH_3)_3CH$	130	75
Propane	C_3H_8	300	210

5.10 *Pressure vessel* shell wall thickness for designated *design pressures* should be in accordance with minimum thickness as required by the U.S.C.G. and Classification Societies for Unfired *Pressure Vessels*.

5.11 *Pressure vessel* head plate thickness for designated *design pressure* should be in accordance with minimum thickness as required by the U.S.C.G. and Classification Societies for Unfired *Pressure Vessels*.

5.12 All *pressure vessels* should be subjected to hydrostatic pressure tests as required by the U.S.C.G. and Classification Societies for Unfired *Pressure Vessels*.

No other test should be substituted for hydrostatic test described above except where specifically *approved* with respect to safety precautions. Pinholes, cracks or other defects should be repaired only by chipping, machining, or burning out the defect and rewelding. Cast iron vessels should not be repaired and if they fail to pass the test should be rejected. Vessels requiring stress relieving should be stress relieved after all welding repairs have been made. After repairs have been made the vessel should again be tested in the regular way. The manufacturer should certify that the welding on the vessel has been done only by welding operators who have passed the test requirements (see paragraph 5.30) and that similar material and technique used in making the tests were employed in fabricating the vessel.

All shell and tube *condenser* water heads should be tested with hydraulic pressure at twice the operating pressure of the system which supplies the water but not less than 50 psi.

Pressure vessels after final hydrostatic test should be leak tested with gas at *design pressure*, cleaned, dehydrated, and sealed.

Design of Condensers, Receivers and Coolers

5.13 Suitable provision should be made for electrical bonding of the various parts of the *condenser* and connected *piping*.

5.14 The tubes should be so arranged that uniform distribution through them would be obtained. Tubes should be expanded, *low temperature brazed*, or welded into the tube sheets, and suitable provision made for difference in expansion coefficient between tubes and shell.

5.15 Minimum tube sheet thickness for rolled tubes should be determined in accordance with the Standards of Tubular Exchanger Manufacturers' Association.

The hardness of the tube sheet should not be less than that of the tubes. Where the tubes are secured to the tube sheet by a method other than the conventional rolling and the staying effect of the tubes has been taken into account in the design, the practice of the manufacturer in determining the thickness of the tube sheet should be specially *approved*.

The required thickness of tubes subject to internal pressure should be determined by the requirements of the U.S.C.G. and Classification Societies for Unfired *Pressure Vessels*.

Table 5-2. Minimum Ligament between Tube Holes

OD tube, in.	*Brazed tubes, in.*	*Rolled tubes, in.*
½	⅛	3/16
⅝	3/16	¼
¾	3/16	¼
1	¼	5/16
1¼	5/16	⅜
1½	5/16	7/16
2	⅜	½

Thinner ligaments may be used where manufacturers have demonstrated this as satisfactory *approved* practice for the *design pressures* and materials used.

5.16 Tube supports should be used where effective tube length exceeds 100 tube diameters. These supports should be placed not more than 48 in. apart for ⅝-in. tubes and 52 in. for ¾-in. tubes or 60 tube diameters for 1-in. and larger tubes, with due consideration given to the possibility of vibration and expansion and of sufficient thickness to prevent cutting of tubes.

5.17 Each *condenser* should be provided with a water drain and air vent in each section of water heads.

5.18 Receivers should be constructed with heads welded or brazed to the shell and so designed and installed as to insure a liquid seal in the receiver outlet under conditions of momentary roll of 30° to either side, and permament list of 15° and permanent trim of 5° fore and aft, and pitch of 5°.

5.19 Each receiver should have a *sight glass* or suitable refrigerant level indicator.

Where *sight glass* is of external type, a self-closing device should be provided.

5.20 The receiver, where used, should have sufficient capacity to take care of refrigerant surges during load changes and to hold the pump-down charge of the largest *evaporator* connected to it where there is more than one, but not less than 50 per cent of the entire refrigerant charge in the system.

5.21 Dished heads may be of circular or elliptical section and may be so placed in the shell that the pressure will be either on the concave or the convex surface.

5.22 Stayed heads may be supported in whole or in part by stays, tubes, ribs, etc.

5.23 The steel plate material used in the construction of heads should be one of the grades suitable for flanging under the general requirements and specifications for shell plates. After heads have been dished, flanged, or locally heated, they should be properly annealed.

5.24 The inside radius of the head formed for attachment to the shell should be not less than three times the thickness of the head and should in no case be less than 6 per cent of the diameter of the shell for unstayed heads. Dished heads having the pressure on the convex side should be formed to a true surface without flats.

The heads here considered are of steel plate. Cast steel heads may be used. The edge of the head should be machined for attaching to the shell.

5.25 Head joints on brazed vessels should have the heads fitted into, or over, the shell with a light driving fit, the overlap being at least equal to 4 times the thickness of the shell plate. See 5.30.

5.26 *Brine coolers* having *brine* in the tubes should follow the recommendations applying to *condensers*.

5.27 *Brine coolers* having *primary refrigerant* in the tubes should follow the recommendations for *condensers* shown in paragraphs 5.13, 5.15, and 5.16 except as follows:

(a) U-tube construction using a single head and removable tube bundle may be used in place of the straight tube and shell construction.

(b) Head to tube sheet joint should be tongue and groove flange type using retained gasket.

(c) Heads should be designed to prevent short circuiting of refrigerant between passes.

5.28 *Water* and *brine coolers* subject to damage through freezing should be fitted with at least two protective devices for the prevention of freezeups.

5.29 Adequate provision should be made for venting, draining, filling, and cleaning of the *brine* side of all coolers.

Welding and Brazing

5.30 *Pressure vessels* may be fabricated by any automatic or manual welding or brazing procedures acceptable to the regulatory body having jurisdiction over the fabrication. *Pressure vessels* to be installed on vessels subject to inspection by the U.S.C.G. or Classification Society are required to be fabricated in accordance with the requirements governing welded fabrication as given in the regulations or rules of the aforementioned regulatory bodies.

5.31 Where heads are bolted on *pressure vessels*, the bolting should be in accordance with the requirements of the U.S.C.G. and the Classification Societies for Unfired *Pressure Vessels*.

Gaskets

5.32 All gaskets should be of materials suitable for use with the *primary refrigerant, brine*, or *cooling water* involved.

Pressure Relief Devices

(See also 6.62-6.66 inclusive)

5.33 Each *pressure vessel* containing liquid refrigerant which may be isolated by valves should be protected by an *approved pressure relief valve*. Where a *pressure vessel* forms an integral part of and cannot be isolated from a system having an *approved pressure relief valve*, such vessel need not have an individual relief valve.

The relief valve, if relieving *high side* pressure, may relieve to the *low side* before relieving to atmosphere. When relieved to atmosphere, further protection against refrigerant loss should be provided by a *rupture disc* in series with the relief valve. Where the relief valve is placed on the *pressure vessel* side, it should be of the type not affected by *back pressure*. A connection should be provided between the *rupture disc* and the relief valve for purposes of determining leakage. A relief valve or *rupture disc* should be fitted on or near each *compressor* on the gas delivery side between *compressors* and before the first *stop valve*.

5.34 *Pressure vessels* where the diameter is larger than 16 in., unless the volume is five cu. ft. or less, should be fitted with two separate *pressure relief valves* in parallel with suitable connections so that one valve is operative at all times. The two relief devices may have common discharge connections and a common connection from the shell.

5.35 If a *low side pressure vessel* can be valved off, the *design pressure* of the *low side* vessel shall govern the setting of the *pressure relief valve* thereon.

5.36 *Rupture discs* and *pressure relief valves* should relieve at a pressure not greater than *design pressure*.

5.37 Sizes of *pressure relief valves* and *rupture discs* should be determined in accordance with paragraph 6.66.

Name Plate

5.38 It is recommended that each *pressure vessel* bear a name plate with the following information:

(a) Type of construction (welded, brazed, etc.)
(b) U.S.C.G., or Classification Society symbol and number assigned to vessel
(c) Manufacturer's name
(d) Serial number
(e) Shop test pressure, psi
(f) *Design pressure*, psi
(g) Month, day, and year of test
(h) Inspector's identification

6. PIPING, VALVES, FITTINGS, AND CONTROLS

6.1 Design, manufacture, and factory test of pipe, flanges, valves, fittings, controls, bolting, and gaskets for *primary refrigerant, brine*, and *cooling water* are included in this section. *Piping* should be understood to comprise all *primary refrigerant, brine*, or *cooling water piping* to be used aboard ship or in factory assemblies intended for such installation.

Materials

6.2 All materials employed should be suitable for handling the *primary refrigerant, brine*, or fluid used. No materials should be used that will unduly deteriorate due to chemical action with *primary refrigerant* or oil, or combination of both. No materials should be used that become unduly brittle at the lowest intended operating temperature. Any combination of metals should be carefully selected by reference to their relative position in the electromotive series in order to minimize electrolytic action.

Design

6.3 Component parts of a *primary refrigerant piping* system should be designed for the minimum *design pressure* shown in Section 5, Table 5-1.

6.4 Pipe wall thicknesses for designated *design pressures* and temperatures should be determined in accordance with requirements of the U.S.C.G. or Classification Society. In no case should the thickness of copper tubing be less than minimum given in Tables 6-2 and 7.3.

General Requirements for Piping

6.5 *Primary refrigerant piping* should be properly protected to prevent damage during normal operations aboard ship. Such protective measures should not interfere with operation.

6.6 The number of breakable joints used in *primary refrigerant piping* should be kept to a minimum by the use of *brazed joints* or welded joints on valves, fittings, sleeves, etc. and by bending tubing or by combining units wherever possible. Wherever *piping* extends through watertight bulkheads, decks, or tank tops, suitable fittings should be installed. The bulkhead plating should not form part of the joint.

6.7 All bolted connections should be threaded through the bulkhead or deck plating and all welds should be on both sides. Whenever steel-to-copper joints are required, they should be flanged or made with suitable adapter fittings *brazed (low temperature)* to the copper pipe and *brazed (high* or *low temperature)* to the steel pipe. Brazing material containing other than residual phosphorus as an impurity should not be used on joints for ferrous material. Slip type expansion joints should not be used. The faces of flanged joints should be parallel before being bolted together. Gaskets should be of material suitable for use with the *primary refrigerant, brine*, or fluid involved. No *soldered joint* should be used.

6.8 Brass, bronze, or copper should not be used when temperatures over 400 F may be encountered.

6.9 All bolts, nuts, studs, pins, screws, cap screws, and other fastenings should be of corrosion resisting material or treated in a suitable manner to render such parts adequately resistant to corrosion. All bolts and nuts should be semifinished, hexagonal shape, and in accordance with dimensions for the Heavy Series of the American

Standard for Wrench Head Bolts and Nuts and Wrench Openings, ASA B18.2. Wherever possible through bolts should be used.

6.10 The inside of all *piping*, valves, fittings, and other apparatus should be clean and free from all foreign matter and injurious defects when installed.

6.11 Minimum centers to which pipe may be bent should be those in Table 6-1.

Table 6-1. Minimum Bending Centers for Pipe or Tubing

Nom size pipe or tubing	*Type K tubing*	*Standard weight pipe*	*Extra heavy pipe*
¼	1¾		
⅜	2½		
½	3	2	2
¾	4¼	2¾	2¾
1	6	3	3
1¼	8	4	3½
1½	10	5	4
2	12	7	6

General Requirements—Valves and Fittings

6.12 Hand valves should be properly labeled as to usage as necessary. Where practicable, all automatic control devices should be mounted immediately outside of the refrigerated compartment which they serve and should be grouped as closely as practicable. Valves, fittings, and controls should bear maker's name or trade mark and reference symbol indicating the service conditions and *design pressure* for which intended.

6.13 All valves should be of such design that the internal working parts may be removed for inspection, repair, or replacement without necessity of breaking line connections. Manually operated valves should be made to close with a right hand (clock-wise) motion of handwheel or operating wrench when facing end of valve stem.

6.14 Automatic control valves should be provided with means for making necessary adjustments externally.

6.15 All wearing parts of valves such as seats, discs, stems, should be of noncorrosive material under service conditions. Only bronze and steel valves, 1½ in. nominal size and smaller should have screwed bonnets and they should be of the union-bonnet type or so constructed as to prevent the bonnet slackening back or loosening when the valve is operated. Valves having stem packings or stuffing boxes should be of the back-seating type to permit repacking under pressure. Valve stems when provided with stuffing boxes should have formed or string packing suitable for intended service and be provided with a suitable gland and means of compressing the packing wherever necessary to secure tightness.

6.16 *Stop valves*, except those used for throttling or expansion purposes, should have an unrestricted flow area at least equivalent to the area of a circle having diameter equal to nominal pipe size. *Stop valves* should be of globe or angle type for *primary refrigerants;* and globe, angle, or gate type for water or *brine* service.

6.17 All gate, globe, or angle valves should be of the rising stem type or specially *approved* type designed to show clearly whether they are in open or closed position.

6.18 Castings should be free from defects. No peening, welding, or soldering should be employed to repair porous spots, sponginess, or cracks, except that defects in steel castings may be welded by an *approved* process after cleaning out to solid metal provided their strength is not impaired. All welds should conform to the requirements of the U.S.C.G. or Classification Society.

6.19 Valves subject to line pressures should be hydrostatically tested by the manufacturer at a pressure not less than two times the recommended *design pressure* and leak tested with gas at a pressure not less than recommended *design pressure.* See Section 5, Table 5-1.

6.20 No valves, *piping*, or fittings should be installed or connections made that would prevent the proper operation of a *pressure relief valve.*

6.21 Automatic control valves should be *by-passed* to permit manual operation or repair in case of failure without discontinuance of operation of system.

6.22 *Stop valves* should be located wherever necessary in addition to the following locations:

(a) Each inlet and each outlet pipe of each reciprocating *compressor*

(b) Each inlet and each outlet pipe of each *liquid receiver.* See Section 5, paragraph 5.33.

(c) Each liquid and each suction branch.

6.23 Necessary and suitable valves, fittings, and connections should be provided to permit *charging*, *purging*, evacuating, draining, venting, and manual operation of system in event of failure of automatic controls.

6.24 Wherever necessary for the proper operation of a system, suitable thermometers and gages should be installed.

6.25 Gaskets should be used on all seal caps.

Solenoid and Electric Motor Operated Valves

6.26 All electrical equipment should conform to A.I.E.E. Recommended Practice for Electrical Installations on Shipboard, latest revision, or Classification Society rule, except as noted herein.

6.27 *Solenoid valves* should be of the packless type of such design that replacement of solenoid coils may be made without breaking conduit or cable connections, and should permit repair or replacement of internal parts without necessity of breaking *piping* connections. A connection box with terminal block should be attached to or incorporated in the coil housing.

6.28 *Solenoid valves* should not be installed within *refrigerated cargo* spaces, or so located as to be subject to the effect of such spaces unless such location is essential for the purpose for which the valves are intended. When so located, the equipment should be watertight.

6.29 All *solenoid valves* should have coil housings of drip-proof construction to exclude falling liquid or solid materials from any angle up to 15° from vertical. All solenoids should have class A insulation (Par. 12.03, Sec. 12, A.I.E.E. Recommended Practice) and should be designed on basis of 40 C temperature rise with 50 C ambient.

6.30 Direct current *solenoid valves* should be provided with suitable means of surge protection. This should either be built into the coil or be located within the coil housing or attached connection box.

6.31 Electric motor operated valves should have a drip-proof operating unit of ample capacity with enclosed gearing and limit switches. Suitable means for operating valves manually should be provided. A suitable drip-proof control panel with indicating lights and line disconnect switch should be provided for each valve unit.

6.32 Solenoids and electric motors should operate satisfactorily with 15 per cent under or over design voltage when valves are subjected to *operating pressure*. All terminals, bolts, nuts, screws, and all parts that would be rendered ineffective by rusting should be of nonferrous material.

Insulation

6.33 All *primary* and *secondary refrigerant piping* should, where required, be adequately insulated.

6.34 *Piping* insulation should be adequately protected against moisture by application of suitable sealing material. Provisions should be made to prevent heat transfer between the cold pipe and any metallic structure by which it is secured or supported. Provision should be made on all pipe hangers to prevent chafing of pipe.

Piping for Freon Refrigerants

6.35 All copper tubing should be seamless drawn with minimum wall thickness conforming to Table 6-2.

Table 6-2. Seamless Drawn Copper Tubing

Nom size	*Actual OD, in.*	*Min wall thickness, in.*
¼	⅜	.049
⅜	½	.049
½	⅝	.049
⅝	¾	.049
¾	⅞	.065
1	1⅛	.065
1¼	1⅜	.065
1½	1⅝	.072
2	2⅛	.083
2½	2⅝	.095
3	3⅛	.109
3½	3⅝	.120
4	4⅛	.134
5	5⅛	.160
6	6⅛	.192

6.36 No *piping* smaller than ¼ in. nominal size should be used in lines where flow of refrigerant is involved except on factory assembled units of ½ horsepower or less.

6.37 Seamless steel *piping* may be used (see paragraph 6.4).

6.38 Pipe should be clean, smooth, round, straight, of proper dimensions, true to form with ends cut square, and should be free of all defects such as seams, grooves, indentations, burrs, cracks, and flaws.

Strainer and Dehydrator for Freon Refrigerants

6.39 A *dehydrator* should be installed in each main liquid line.

6.40 A *strainer* of ample size and proper mesh screen of noncorroding material as given in Table 6-3 should be installed in liquid lines ahead of each *solenoid* or *thermostatic expansion valve* or combination of such and in all main suction lines. The size wire used in construction of *strainers* should be such that the open area will be at least 30 per cent of the total screen area.

Table 6-3

Size line actual OD, in.	*Nom size, in.*	*Liquid lines min screen area Max mesh*	*Liquid lines min screen area Sq in.*	*Suction lines min screen area Max mesh*	*Suction lines min screen area Sq in.*
¼	⅛	100	2.5	—	—
⅜	¼	100	5	50	5
⅝	½	100	11	50	11
⅞	¾	100	20	50	20
1⅛	1	100	23	50	23
1⅜	1¼	100	30	50	30
1⅝	1½	100	40	50	40
2⅛	2	100	50	50	50
2⅝	2½	100	60	50	60
3⅛	3	100	70	50	70
4⅛	4	100	80	50	80
5⅛	5	100	90	50	90
6⅛	6	100	100	50	100

6.41 *Dehydrators* and *strainers* should be so constructed that the drying agent or screen may be removed and replaced without breaking *primary refrigerant* line connections. The head connections of the shells of *dehydrators* and *strainers* should be of the tongue and groove flanged type.

6.42 A *by-pass* should be provided around all *strainers* and *dehydrators* so that cleaning and replacement may be accomplished without interrupting operation.

Valves and Fittings for Freon Refrigerants

6.43 *Stop valves* should be of the packless diaphragm type in ¾ in. nominal size and smaller. Larger sizes may be of the packed rising stem type provided seal caps are used. No compression fittings (flare) larger than ¼ in. should be used except on factory assembled units where such fittings may be used in sizes up to ⅝ in. inclusive, provided joints are exposed for visual inspection.

6.44 All valves and fittings using *brazed joint (low temperature)* type connections should be provided with minimum depth pipe sockets in accordance with Table 6-4. Clearances between outside diameter of tube and inside diameter of socket should not be less than 0.002 in. nor more than 0.006 in. All valve parts subject to damage should be removed during brazing operations.

Table 6-4

Nom size, in.	*Actual OD, in.*	*Min depth of socket, in.*
⅛	¼	1/4
¼	⅜	17/64
⅜	½	9/32
½	⅝	5/16
¾	⅞	3/8
1	1⅛	13/32
1¼	1⅜	7/16
1½	1⅝	1/2
2	2⅛	5/8
2½	2⅝	11/16
3	3⅛	13/16
3½	3⅝	7/8
4	4⅛	15/16
5	5⅛	1-1/16
6	6⅛	1-1/8

6.45 All fittings in sizes up to one inch nominal inclusive should be of wrought copper or other *approved* material.

6.46 Valves should be of forged brass, forged steel, high tensile cast iron, or cast bronze in sizes up to and including one inch nominal. For larger sizes, high tensile cast iron, forged or cast steel may be used provided the valves are of the flanged retained gasket type with brass adapter fittings or weld neck companion flanges.

6.47 All copper connections larger than ¼ in. nominal size should be of the *brazed joint (low temperature)* type except as noted in paragraphs 6.7 and 6.36.

Piping, Valves and Fittings—Carbon Dioxide
(Design Pressure 1500 psi)

6.48 All *piping* should be of seamless steel or copper.

6.49 Fittings should be of forged or cast steel.

6.50 Wherever possible all connections should be of the welded type. Valves should be of forged or cast steel with bronze or stainless steel working parts.

6.51 Wherever flanged connections are necessary they should be of the retained gasket or annealed copper ring type.

Pipe, Valves and Fittings—Electrolytic Brine
(Design Pressure 125 psi)

6.52 Pipe should be of steel, wrought iron, or copper.

6.53 No galvanizing should be applied to any part of any pipe, valve, or fitting in contact with *electrolytic brine.*

6.54 Screwed joints may be used in sizes up to 1½ in. and flanged or welded joints may be used in all sizes. For copper pipe, *brazed (low temperature)* bronze unions may be used in sizes up to 1½ inches and *brazed (low temperature)* bronze flanges may be used for all sizes.

6.55 Fittings should be 150 lb standard steel or 125 lb cast iron or 150 lb malleable or wrought iron. For copper pipe with *brazed joints (low temperature)*, brazed bronze fittings may be used.

6.56 Valves should be of bronze or cast iron with bronze working parts. See paragraphs 6.2 and 6.17.

6.57 No bronze valves with screwed connections should be used with ferrous *piping.*

Pipe, Valves and Fittings—Ammonia
(Design Pressure 300 psi)

6.58 Steel pipe or tubing should be used.

6.59 Connections may be screwed, welded, or flanged (retained gasket type). Screwed type connections should not be used in sizes 1½ in. and larger.

6.60 Flanges and fittings should be of cast or forged steel, or malleable or cast iron.

6.61 Valves should be of rising stem globe or angle type and of cast iron, forged steel, or cast steel. (See 6.2 and 6.17.) Cast iron valves and fittings should be of flanged retained gasket type.

Pressure Relief Devices
(See also 5.33-5.37 inclusive)

6.62 All *pressure relief valves* should be clearly marked with the following data:
- (a) Name or trade mark of manufacturer
- (b) Serial number
- (c) Relieving pressure and discharge capacity to atmosphere (lb air per min)

6.63 No *stop valves* should be located between a *pressure relief valve* and the part of the *refrigerating system* protected by such relief valve unless two *pressure relief valves* are provided and arranged so that only one can be shut off at a time.

6.64 All *pressure relief valves* should be set to relieve at a pressure not greater than the *design pressure* of the system. All *pressure relief devices* should be connected directly, or as near as practicable, to the *pressure vessel* or other parts of the system protected thereby, and should be placed above the liquid refrigerant level. All *pressure relief devices* should be directly actuated by the *primary refrigerant* pressure.

6.65 *Rupture discs* should be clearly stamped with manufacturer's name, material, bursting pressure, and temperature.

NOTE: It should be noted that bursting discs smaller than 1 in. dia. have a tendency to be erratic in operation for bursting pressures below 250 psi. For this reason it is recommended that for bursting pressures below 250 psi the bursting disc should be not less than 1 in. dia.

6.66 The minimum required rated discharge capacity of the *pressure relief device* for a *pressure vessel* should be determined by Formula in the A.S.A. B9 Code.

Controls

6.67 All parts of control mechanisms subject to refrigerant pressure should have a factory test pressure of not less than 25 per cent above the *design pressure.*

6.68 Electrical features of control mechanisms should conform to latest revision of A.I.E.E. Recommended Practice for Electrical Installations on Shipboard.

6.69 Control mechanisms or parts thereof that may suffer functional damage if subjected to required system test pressure after installation should be isolated during test.

6.70 Controls should be so designed that their proper operation will not be affected by ship motion.

6.71 No *stop valve* should be installed between any *pressure limiting device* and the part of the system which it protects.

Gaskets

6.72 All gaskets should be of materials suitable for use with the *primary refrigerant, brine,* or *cooling water* involved.

6.73 Rubber gaskets should not be used for *primary refrigerants.* No gasket materials should be used that will change in character so as not to perform the service intended at the temperatures involved. No gasket material should be used that will deteriorate due to contact with the fluid involved.

7. EVAPORATORS AND COOLING COILS*

7.1 For the purpose of these Recommendations, *low sides* and *cooling coils* will be considered: first, for *direct expansion systems;* and second, for *indirect systems.*

7.2 Where ammonia is the *primary refrigerant,* only steel or iron pipe should be used for *cooling coils.*

7.3 The tube or pipe wall thickness of *evaporators* and *cooling coils* should be determined in accordance with requirements of U.S.C.G. or Classification Society but in no case be less than the following minimum requirements:

---

* Does not apply to *low sides* and *cooling coils* for carbon dioxide refrigerant.

size OD, in.	Min wall thickness, in.	
	Copper	Steel
3/8	.025	.042
1/2	.025	.042
5/8	.025	.049
3/4	.035	.049
7/8	.035	.065
1 1/8	.065	.072
1 3/8	.065	.072
1 5/8	.072	.095

Where pipe is used, it should be at least full standard weight pipe. Return bends should have at least the same wall thickness as tubes.

7.4 Where fins are used, the material should be copper, aluminum, steel, or other suitable material.

7.5 The tube sheets supporting the tubes should be copper, steel, or other suitable material.

7.6 The tube sheets supporting the tubes in the coils should be of sufficient thickness and so supported as to provide a rigid coil assembly.

7.7 All hangers and supporting members should be of sufficient strength to absorb stress due to normal pitching, rolling, and vibration.

7.8 All joints on copper tube coils should be preferably *brazed joints (low temperature)*. All joints on steel tube coils should preferably be welded.

7.9 Copper coils should be coated with tin or other suitable material by either hot dipping, electroplating, or other *approved* process.

7.10 Steel coils should be hot dipped galvanized after assembly except as noted in paragraph 7.33.

7.11 Unit cooler casings should have an *approved* protective coating.

7.12 Hangers and supporting members should be protected against corrosion.

7.13 Steel fan rotors should be zinc coated, hot dipped, galvanized, or have other *approved* protective coating.

7.14 All coils, after assembly, should be tested with gas or air pressure while under water at 1½ times the *design pressure*, as listed in Table 5-1.

7.15 Adequate means should be provided for defrosting blower coils dependent upon fin spacing, fin depth, and conditions of service.

7.16 All drain pans on blower units should be insulated where damage to cargo from sweating may occur. Minimum drain size should be ¾ in. pipe but, in all cases, should be ample for draining and *defrosting.*

7.17 All drain traps should be installed, whenever possible, outside refrigerated space.

7.18 The motor and fan mounting on ceiling blower units should be attached direct to a solid foundation with suitable damping arrangements, to avoid vibration troubles.

7.19 The casings of blower coils should be arranged for easy access to coils, fans, motors, and valves.

7.20 All fans with belted drives should be screened or guarded to prevent injury to operating personnel.

7.21 All coils should be rated in accordance with ASRE Standard #25 (latest edition).

7.22 Where necessary, suitable means for defrosting should be provided.

7.23 For hot gas *defrosting* the necessary valves and devices should be provided for *by-passing* the expansion valves and orifices.

7.24 For electric *defrosting,* a high-limit switch should be provided to prevent possible overheating.

7.25 The motors used on fan *cooling coil* units should be drip-proof and insulated in accordance with latest revision, A.I.E.E. Recommended Practice for Electrical Installations on Shipboard.

7.26 Motor selection and application should take into account the desirability of speed control to provide flexibility in operating conditions.

7.27 Fans and motors for use with low temperature coils should be selected with sufficient capacity to allow for increased static pressure due to frost accumulation.

7.28 When *heat interchangers* are used, provisions should be made to prevent oil trapping on the suction side.

7.29 Construction of *scuttlebutts* should conform to recommendations in Section 5 and latest regulations of the U. S. Public Health Service.

7.30 *Scuttlebutts* should be provided with suitable devices to prevent freeze-ups.

7.31 Wrought iron, steel, or copper tube or pipe used in *cooling coils* for *indirect systems* should have a minimum wall thickness in accordance with paragraph 7.3.

7.32 Where copper is used in *cooling coils* for *indirect systems,* the *brine* should be maintained alkaline at all times.

7.33 Care should be taken that calcium *brine* does not come in contact with any galvanized surface.

7.34 *Brine coolers* used with *electrolytic brine* should not be constructed with dissimilar metals, unless precaution is taken to prevent electrolytic action.

7.35 Thickness of tubes in *brine coolers* should be in accordance with minimum thickness as required by the U.S.C.G. and Classification Societies.

7.36 Tube sheet thickness should be in accordance with recommendations in Section 5.

7.37 The heads and shell should conform to standard *pressure vessel* design, as recommended in Section 5.

7.38 Provision should be made for venting and draining the *brine* side of coolers or *evaporators.*

7.39 Provision should be made for draining refrigerant side of coolers or *evaporators.*

7.40 When *electrolytic brine* is used with steel or wrought iron *piping,* the *brine* pumps should have ferrous casings.

7.41 For *low side* pressure relief requirements, see paragraphs 5.33, 5.34, and 5.35.

8. INSTALLATION AND TESTS

8.1 It should be established that:

(a) All *pressure vessels* and *compressors* have been strength and leak tested at the point of manufacture.

(b) All *piping,* valves, fittings, and controls have been manufactured and tested at the point of manufacture in accordance with Section 6.

(c) All *cooling coils* and *low side* equipment have been manufactured and tested at the point of manufacture in accordance with Section 7.

No test should be substituted in lieu of the above. Where there is no positive evidence that these tests have been performed at the point of manufacture, the items involved should be returned to the point of manufacture for testing or suitable equivalent hydrostatic tests performed using a liquid having an atmospheric boiling point and flash point above 150 F and in the presence of qualified inspectors, before any gas test is applied. It is recommended that water not be used to test hydrostatically *primary refrigerant* piping, valves, fittings, or equipment.

It is the intention of the above to prevent damage due to bursting of defective *pressure vessels* or parts under gas pressure and to provide for all strength testing of parts before assembly and to confine testing of the completely installed *primary refrigerant* system to that necessary to insure tightness against refrigerant leaks.

8.2 The completely installed and assembled *primary refrigerant* system should be tested under gas pressure to the applicable values listed in Table 8-1, and in accordance with the procedure described in paragraphs 8.4 to 8.15 inclusive.

8.3 Where the refrigerant used is not listed in Table 8-1, the *low side* test pressure should be equal to the saturated vapor pressure of the refrigerant at 115 F and that of the *high side* should be equal to the saturated vapor pressure of the refrigerant at 150 F. Test pressures given in Table 8-1 are equal to *design pressure* given in Table 5-1. It is intended that *pressure relief valves* and other safety devices remain connected to the system during testing except for *rupture discs*. (See paragraph 8.12.)

8.4 In no case, for any refrigerant, should the test pressure be less than 30 psi.

8.5 It is recommended that oil pumped dry nitrogen or bone dry carbon dioxide with a detectable amount of the intended refrigerant added, be used as a testing medium. In no case should oxygen or any flammable gas or flammable mixture of gases be used for testing.

NOTE: The above tests are intended to be applied to new installations. Carbon dioxide should not be used to leak test systems using ammonia as a *primary refrigerant* except for leak test prior to charging.

8.6 Gas testing medium should be admitted to the system only through a reliable pressure regulating device which will prevent the pressure from rising beyond safe values.

8.7 Test pressure should be indicated on at least two gages or other pressure measuring instruments observed by two operators, each of whom should be within easy reach of a provision to reduce or relieve the test pressure.

8.8 On any *primary refrigerant* system containing a *low side pressure vessel* in accordance with Section 5, such *pressure vessel*, all suction *piping*, and liquid *piping* up to and including the nearest *stop valve* should be tested to the applicable value for *low sides* given in Table 8-1. The remainder of the system should be tested to the applicable value for *high sides*, Table 8-1, excluding the compressor body and any gages, control mechanisms, or parts thereof that may suffer functional damage if subjected to required system test pressure.

8.9 In any *primary refrigerant* system not containing a *low side pressure vessel* in accordance with Section 5 the entire system with the exclusions stated in paragraph 8.8 above should be tested to the applicable value for *high sides*, Table 8-1.

8.10 Valves which have remained closed to protect compressor bodies from *high side* test pressures as described in paragraphs 8.8 and 8.9 above should be opened after the pressure has been reduced to the applicable value for *low sides* to detect any possible leaks in the shaft seal, gaskets, *piping* connections, or elsewhere.

8.11 Valves that have remained closed to protect low range pressure gages or control mechanisms from excessive pressure as described under paragraphs 8.8 and 8.9 above should be opened after the pressure has been reduced to a value not less than one half of the gage scale range or control mechanism operating range, to detect any possible leaks in the unit or the *piping* connections.

8.12 It is the intention in the above paragraphs that all parts to be subjected to *primary refrigerant* pressure during operation be completely connected to the system before testing and that no such parts except *rupture discs* be removed from the system for replacing after the test. Segregation of test pressures between the *high side* and *low side* of the system should be accomplished by the use of suitable permanently in-

Table 8-1. Test Pressures after Installation.

Refrigerant	*Formula*	*High side, psi*	*Low side, psi*
Group 1			
Carbon Dioxide	CO_2	1500	1000
Dichloromethane (Methylene Chloride)	CH_2Cl_2	30	30
Freon-11 (Trichloromonofluoromethane)	CCl_3F	30	30
Freon-12 (Dichlorodifluoromethane)	CCl_2F_2	225	150
Freon-21 (Dichloromonofluoromethane	$CHCl_2F$	70	40
Freon-22 (Monochlorodifluoromethane)			
Water Cooled *Condensers* Only	$CHClF_2$	300	150
Freon-113 (Trichlorotrifluoroethane)	$C_2Cl_3F_3$	30	30
Freon-114 (Dichlorotetrafluoroethane)	$C_2Cl_2F_4$	50	50
Group 2			
Ammonia	NH_3	300	150
Dichloroethylene	$C_2H_2Cl_2$	30	30
Ethyl Chloride	C_2H_5Cl	60	50
Methyl Chloride	CH_3Cl	210	125
Methyl Formate	$HCOOCH_3$	50	50
Sulphur Dioxide	SO_2	170	95
Group 3			
Butane	C_4H_{10}	95	50
Ethane	C_2H_6	1200	700
Isobutane	$(CH_3)_3CH$	130	75
Propane	C_3H_8	300	210

NOTE: *No gas test should be made aboard a vessel at any time to pressures higher than those listed above.*

stalled *stop valves* or suitable blank flanges.

8.13 The initial test of a newly installed system should be to establish the ability of the system to hold gas pressure by observation of the test gage. If the refrigerant used is adaptable to the use of some form of *leak detector*, the system should then be relieved to atmosphere and sufficient refrigerant introduced to raise the pressure to about 10 psi. Flame type detectors should not be used where the refrigerant used is flammable. The test pressure should be brought up to the required value in several steps using inert gas as described in paragraph 8.5 and the *leak detector* should be used after each step. Tightness of shell and tube *condensers* should be determined by removing the heads from the water side and applying the *leak detector*.

8.14 Where a leak is discovered which will require application of heat to repair, the part of the system containing the leak should be opened to atmosphere and all test gas relieved to a point outside of the ship's enclosure so as to prevent any health hazard due to deficiency of oxygen.

8.15 After the system has been completely tested and proved leak tight in accordance with the above, all test gas should be relieved to the atmosphere as described in paragraph 8.14 and *rupture discs* replaced. The system should then be dehydrated, charged, and otherwise prepared for initial operation as described in Section 10. An additional and final check on the tightness of the system should be made after the dehydration procedure by noting whether it will hold a vacuum of not less than 26 inches for a period of six hours.

8.16 *Brine* systems should be tested hydrostatically in accordance with the latest rules of the Classification Societies and U. S. Coast Guard.

9. SPARE PARTS AND TOOLS

General

9.1 The quantity and types of spare parts for the maintenance of refrigeration equipment at sea that should be carried by ships to safeguard their *refrigerated cargoes* or *ships stores* should be based upon the amount of stand-by equipment carried by the ships.

9.2 It should not be necessary to carry a full complement of spare parts for the mechanical equipment of any system in which the required equipment as designed is duplicated with 100 per cent spare condensing units, and both the required and duplicate installations meet all of the following requirements:

(a) The refrigerant *piping* is arranged for complete interconnection and fully provided with isolating valves.

(b) Each of the installations is capable of adequately dealing with *refrigerated cargo* or refrigerated *ship's stores* as received on board, and with one condensing unit in reserve the remaining unit or units are capable of maintaining the required temperature in the refrigerated spaces when in tropical climates with the machine running 24 hours a day.

(c) All similar parts are interchangeable.

Packaging and Labeling

9.3 All spare parts should be suitably packed and protected against humidity and salt atmosphere, individually labeled, and stowed for protection against mechanical damage.

Miscellaneous

9.4 Regardless of the amount of stand-by equipment all installations should include essential accessories, special tools, spare charges, and certain vital parts for the *compressors*, not less than the list below, where applicable.

NOTE: Spare parts should be furnished in accordance with the following list as applicable on the basis of the total equipment supplied by each manufacturer for any one vessel.

9.4.1 ***General***

10 percent spare tubes of each size for one *condenser* and one *brine cooler (evaporator)*

200 percent spare *condenser* head gaskets for one *condenser*

Spare *compressor* gaskets and shaft or rod packing or shaft seal assemblies of sufficient quantity to service not less than 25 per cent of each group of similar *compressors* comprising the installation

Set of discharge and suction valves for one *compressor* of each similar type

One thermometer and one pressure gage each size, type, and scale range for each ten or fraction duplicate instruments installed.

9.4.2 ***Accessories***

One vacuum pump (not necessary on installations having 25 total connected *compressor* horsepower or less)

One flexible charging connection

One *oil charging pump*, including charging line

One *leak detector*

9.4.3 ***Special Tools***

One tool kit to include all special tools required to service any of the refrigerating machinery or apparatus

One tube expander tool or beller tool for *condensers* and *brine cooler (evaporators)*

9.4.4 ***Spare Charges***

One complete charge of refrigerant for the equipment required for normal operation

Consideration should be given to the amount of basic materials necessary for replenishing the *brine* in an *indirect system*, depending upon the type of service

One complete charge of *compressor* lubricating oil, the spare charge to be of sufficient quantity to renew the oil in the entire system

Spare charge for each *dehydrator*

9.4.5 ***Refrigerant Line Fittings and Accessories***

	Quantity duplicate installed units			
	2-5	*6-8*	*9-11*	*12 & Over*
Necessary items	*Quantity spare parts*			
Cartridge assemblies for *dehydrators*, complete	See 9.4.4 above			
Sight glass for *liquid receivers*, sets, complete with gaskets	1	1	1	2
Rupture discs	100% spares			
Pressure relief valves	1	1	2	2
Solenoid valve coil	2	3	3	3
Strainer baskets	1	1	2	2
Gaskets for *strainers*, sets	2	2	4	4
Thermostats	1	1	2	2
Hand expansion valve	1	1	2	2
"Cage" assembly for *thermostatic expansion valve*	1	2	3	4

9.4.5 *(Cont'd)*

	Quantity duplicate installed units			
	2-5	*6-8*	*9-11*	*12 & Over*
Necessary items	*Quantity spare parts*			
Power assembly for *thermostatic expansion valve*	1	2	3	4
Liquid distributor assembly, complete except for lower body	1	1	2	2
Packless refrigerant *stop valve*	1	1	2	2
Complete valve assemblies less lower bodies for packed-type *stop valves*	1	1	2	2
Packing and gaskets for "seal-cap" type *stop valve*	1	2	3	4
Evaporator pressure regulator, complete except for lower body	1	1	2	2
Float valve	1	1	1	1

9.4.6 *Fans*

	Quantity duplicate installed units	
	2-8	*9 & over*
Necessary items	*Quantity spare parts*	
Coupling bolts and bushings, sets	1	2

9.4.7 *Brine Cooler Type Refrigerant Evaporators and Heat Interchangers, Brine Heaters*

	Quantity duplicate installed units		
	2-3	*4-5*	*6 & over*
Necessary items	*Quantity spare parts*		
Tubes	See 9.4.1 (a) above		
Gaskets, total all units	100%	100%	100%

9.4.8 *Electrical Equipment.*

(See par. 9.12 to 9.19 incl.)

Condensing Units

9.5 The following spare parts, where applicable, should be furnished in addition to the minimum requirements given in paragraph 9.4. Duplicate standby units in excess of those required by paragraph 9.2 (b) may be considered as spares. The spares listed in paragraphs 9.6 to 9.11, inclusive, need not be carried if the plant is for *ship's stores* only and the voyage is of three days or less in duration.

9.6 *Freon Reciprocating Compressors*

	Quantity similar compressor units		
	2-5	*6-8*	*9-12*
Necessary items	*Quantity spare parts*		
Shaft packing, sets complete	1	2	3
Crank shaft, complete with main bearings	1	1	2
Cylinder liner	50% for one *compressor* unit, but not less than 2		

9.6 *(Cont'd)*

	Quantity similar compressor units		
	2-5	*6-8*	*9-12*
Necessary items	*Quantity spare parts*		
Piston, complete with rings and piston pin and suction valve	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit		
Piston rings	50% for one *compressor* unit, but not less than total of 3 piston sets where more than one *compressor* unit		
Valve plate assembly, complete	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit		
Connecting rod, complete with bushings, bearings, and bolts	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit		
Unloading assembly, complete	1	1	2
Oil pump, complete with *strainers*	1	1	2
Basket for suction *strainer*	1	1	2
Suction and discharge *stop valves,* complete including packing, less bodies	1	1	2
Compressor relief valve assembly, complete	1	1	2
Oil *sight glass* and gaskets	1	1	2
High and low pressure cut-out switch	1	1	2
Float valve assembly, complete	1	1	2

9.7 *Ammonia Compressors*

	Quantity similar compressor units	
	2-5	*6 & over*
Necessary items	*Quantity spare parts*	
Shaft packing, sets complete	1	2
Lantern gland	1	1
Pump-out and starting *by-pass* valves, with packings, less bodies	1	1
Crankshaft, complete with main bearings	1	1
Cylinder liner	50% for one *compressor* unit, but not less than 2	
Piston, complete with rings and piston pins and suction valve	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit	
Piston rings	50% for one *compressor* unit, but not less than total of 3 piston sets where more than one *compressor* unit	
Valve plate assembly, complete	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit	
Connecting rod, complete with bushings, bearings, and bolts	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit	
Unloading assembly, complete	1	1
Oil pump, complete with *strainers*	1	1
Basket for suction *strainer*	1	1
Suction and discharge *stop valves,* complete including packing, less bodies	1	1
Compressor relief valve assemblies, complete	1	1
Oil *sight glass* and gaskets	1	1

9.7 *(Cont'd)*

	Quantity duplicate installed units	
	2-5	*6 & over*
Necessary items	*Quantity spare parts*	
High and low pressure cut-out switch	1	1

9.8 *Carbon Dioxide Compressors*

Shaft packing, sets complete	1	2
Lantern gland	1	1
Crankshaft, complete with main bearings	1	1
Cylinder liner	50% for one *compressor* unit, but not less than 2	
Piston, complete with rings and piston pins and suction valves	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit	
Piston rings	50% for one *compressor* unit, but not less than total of 3 piston sets where more than one *compressor* unit	
Suction and discharge valves, complete	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit	
Connecting rod, complete with bushings, bearings and bolts	50% for one *compressor* unit, but not less than total of 3 where more than one *compressor* unit	
Oil pump, complete with *strainers*	1	1
Basket for suction *strainers*	1	1
Suction and discharge *stop valves*, complete including packing, less bodies	1	1
Compressor relief valve assemblies, complete	1	1

9.9 *Centrifugal Compressors*

	Quantity similar compressor units	
	2-3	*4 & over*
Necessary items	*Quantity spare parts*	
Shaft seal assembly, complete with gaskets	1	2
Shaft labyrinth packing	1	2
Oil pump, complete	1	1
Thrust bearing, complete	1	1
Complete set gaskets	1	1
Testing shield (to protect shaft seal)	1	1
Cartridge for oil *filter*	1	1
Bearings	1	1
Float valve assembly, complete	1	1
Complete set of spare parts for purge unit as specified for reciprocating *compressors*, motors, and drives elsewhere		

9.10 *Condensers*

	Quantity similar condensers	
	2-8	*9-12*
Necessary items	*Quantity spare parts*	
Spare tubes for one *condenser*	10%	20%
Gaskets, sets complete	100%	100%

9.11 *Pumps*

	Quantity Similar pumps		
	2-5	*6-11*	*12 & over*
Necessary items	*Quantity spare parts*		
Impeller, complete with shaft bearings, seal, and shaft sleeve	1 for each size and type pump		
Shaft packing, sets, for one pump of each type and size	1	2	3
Wearing ring, sets, each size and type pump	1	2	3
Sets of suction and delivery valves	1 for each size and type pump		

Driving Units

9.12 *Drives*

	Quantity similar drive units		
	2-5	*6-11*	*12 & over*
Necessary items	*Quantity spare parts*		
Set of belts	1	2	3
Coupling assembly, complete	1	2	3

9.13 *Squirrel-Cage AC Motors*

Complete sets of stator coils and slot insulation	1	2	3
Rotor	1	2	3
Set of bearings	1	2	3

9.14 *Wound Rotor AC Motors*

Complete set of stator coils and slot insulation	1	2	3
Rotor	1	2	3
Set of rotor coils and slot insulation	1	2	3

	Quantity similar drive units		
	2-5	*6-11*	*12 & over*
Necessary items	*Quantity spare parts*		
Brush holder with stud insulation and 3 springs	1	2	3
Set of brushes	1	2	3
Set of bearings, complete	1	2	3

9.15 *Synchronous AC Motors*

Complete set of stator coils and slot insulation	1	2	3
Rotor	1	2	3
Set of rotor coils and slot insulation	1	2	3
Brush holder with stud insulation and 3 spare springs	1	2	3
Set of brushes	1	2	3
Field coil	1	2	3
Set of bearings, complete	1	2	3
Exciter spares	1	2	3

9.16 *DC Motors*

Armature, complete with commutator and shaft	1	2	3
Field coil of each type	1	2	3
Complete brush holder, with stud insulation, 3 spare springs	1	2	3
Set of bearings, complete	1	2	3
Set of brushes	1	2	3

9.17 *Magnetic Motor Controllers*

Set of arc-rupturing and interlock contacts and springs for each circuit breaker, contactor, relay, etc.	1	2	3
Operating and shunt coil, each size and type installed	1	2	3

9.18 *Manual Motor Controllers*

Set of fingers, contacts, and springs	1	2	3
Resistor of each capacity used	1	2	3
Push button station, each type	1	2	3
Fuses	1	2	3

9.19 *Panel Boards*

9.19.1 *Fuses*—Two fuses should be supplied for each fuse installed, up to a maximum of 30 spare fuses for each rating.

9.19.2 *Air Circuit Breakers and Contactors* — For each of the following there should be supplied identical parts equal to the number installed, up to 6; above the first 6 there should be supplied one part for every group of 10 identical parts installed, or a fraction thereof:

Contacts (arc) moving and stationary.

Contacts (auxiliary switch) moving and stationary where renewable types are installed.

Switches (auxiliary) complete, where nonrenewable contacts are installed.

Springs, complete, including those required in pole units, mechanism, and attachments.

NOTE: For coils see Voltage Coils.

9.19.3 *Moulded Case, Thermal Type, Air Circuit Breakers*— One complete trip element assembly, where interchangeable elements are used, for every group of 10 identical trip element assemblies installed, or for a fraction thereof. One complete air circuit breaker, where noninterchangeable trip elements are used, for every group of 10 identical breakers installed, or a fraction thereof.

9.19.4 *Protective and Control Relays*— One complete induction type protective relay should be supplied for every group of 10 identical relays installed or for a fraction thereof.

For control (plunger or armature type) relays there should be supplied one of the following parts, where applicable, for every group of 10 identical parts, or for a fraction thereof:

Contacts, stationary and moving.

Springs.

Coils, blowout.

9.19.5 *Voltage Coils*— One voltage coil of each rating and type installed should be supplied. This should include all voltage coils such as operating mechanism coils, shunt trip coils, voltage coils for reverse current and undervoltage devices, and protective relay voltage coils.

For breaker closing coils using copper oxide rectifiers, the rectifiers should be included with the coil.

For breaker closing motors, there should be supplied one motor of each size and type installed.

9.19.6 *Rotary Type Control and Instrument Switches*— One of each of the following should be supplied for every identical group of 10 parts installed, or for a fraction thereof:

Moving contact segment, if required.

Contact fingers, stationary or movable as applicable.

Spring centering.

9.19.7 *Indicating Lights*— One lens for each group of 10 similar lenses installed or for a fraction thereof. Two lamp bulbs for each lamp installed.

9.19.8 *Resistors*— One resistor unit for every group of 4 identical units installed, up to a maximum of 4 similar units.

9.19.9 *Rheostat*— One rheostat plate of each type and rating installed.

9.20 *Steam Turbines*

	Quantity duplicate drive units	
	2-5	*6 & over*
Necessary items	*Quantity spare parts*	
Set of bearings	1	1
Gland packing, with springs for glands, complete sets	1	2
Emergency governor springs, sets	1	2
Main governor springs, sets	1	2
Main governor wearing parts, sets	1	2
Governor valve stem and seat, complete	1	2
Oil pump	1	2
Relief valve for oil system	1	2
Thrust-bearing, complete	1	2
Spare parts for steam *condenser* as specified for *condensers*	See 9.10	

9.21 *Reciprocating Steam Engines*

	Quantity similar drive units		
	2-5	*6-11*	*12 & over*
Necessary items	*Quantity spare parts*		
Main bearings, sets	1	2	3
Connecting rod bearings, sets	1	2	3
Valve stem and nut of each pattern	1	2	3
Piston rings, complete sets	1	2	3
Piston valve, complete with rings	1	1	2
Eccentric strap and rod, complete with bolts and nuts	1	1	2
Governor springs, sets	1	2	3
Packing, sets	1	2	3
Bolts and screws for main bearings and connecting rod bearings, sets	1	1	2

9.22 *Diesel Engines*

Piston, complete with connecting rod	50 per cent for one engine, but not less than 2
Piston rings, sets complete	100 per cent for one engine
Piston pin bushing and dowel	50 per cent for one engine
Cylinder head, complete	50 per cent for one engine
Injection valves, complete	100 per cent for one engine
Inlet valves, complete	100 per cent for one engine
Exhaust valves, complete	100 per cent for one engine
Main bearings	100 per cent for one engine
Connecting rod bearings	100 per cent for one engine
Cam rollers, set	100 per cent for one engine
Springs, set	100 per cent for one engine
Fuel pump packing, sets complete	100 per cent for one engine
Gaskets, complete sets	100 per cent for one engine
Injection valve packing, sets	100 per cent for one engine
Injector valve spindles	100 per cent for one engine
Injector valve seats or burning plates	100 per cent for one engine

22 *(Cont'd)*

jector pump	100 per cent for one engine
ain governor springs	100 per cent for one engine
ain governor wearing parts	100 per cent for one engine
uds and nuts for one cylinder cover	100 per cent

10. OPERATION, MAINTENANCE, AND INSPECTION

0.1 This section deals with the operation, maintenance, and inspection of mechanical refrigeration equipment installed aboard merchant vessels. The recommendations set forth are considered minimum requirements, and are not intended to replace the inspection requirements of the various marine regulatory bodies such as the American Bureau of Shipping and the U. S. Coast Guard.

Operating Instruction Manuals

10.2 Manufacturers supplying mechanical refrigeration equipment for shipboard application should furnish for each installation a set of instructions for the operation, maintenance, and inspection of the equipment. These instructions should adequately cover all of the necessary information pertaining to the installation, operation, maintenance, and inspection of the equipment and should include at least the information specified in paragraphs 10.3 to 10.8.

10.3 ***Introduction.*** This chapter should include a general description of the equipment, the refrigerant, and the type of refrigeration cycle. A brief section on the principles and fundamentals of refrigeration should also be provided. A table of the thermodynamic properties of the refrigerant should also be included.

10.4 ***Description of Refrigeration Plant.*** This chapter should contain a complete detailed description of the component assemblies and accessories which comprise the complete installation. A complete list of all pertinent equipment and their functions should be given.

10.5 ***Principles of Operation.*** This chapter should provide a brief resume of the principles of operation of the system, or the components of the system, together with such illustrations, sketches, and diagrams as are necessary.

10.6 ***Installation Instructions.*** This chapter should provide complete instructions for the proper installation of the system. It should contain adequate information for the proper dehydration, leak testing, and pressure testing of the plant. Sufficient emphasis should be made of the need to keep the system internally clean.

10.7 ***Operating Instructions.*** This chapter should be prepared for the specific installation and equipment involved. It should cover complete instructions for the operation of the equipment including precautions which should be made before initial starting after installation or after a major repair. Instructions should be included for the charging of refrigerant, oil (and *brine* for *indirect system*), and should also include the instructions for removing the refrigerant and oil from the system. The type of *leak detector* suitable for the plant should be stated and explanation of use should be given. An adequate description should be given for the proper adjustment of all controls, including safety devices. Where applicable, instructions should be provided for the *defrosting* of *cooling coils*. Operation difficulties should be explained together with their corrective measures. This may be presented in the form of a Trouble Diagnosis Chart. Instructions should be given for the periodic check of temperatures and pressures. The design holding temperatures of all *ships stores* and cargo spaces should be provided. A listing of safety precautions should be included; such as, in no case should air, oxygen, or flammable mixture of gases be used for testing. First aid treatment for injuries resulting from escaping refrigerant should be included.

10.8 ***Maintenance.*** This chapter should provide all the necessary instructions for the proper care and maintenance of the equipment. This should include instructions for the inspection, adjusting, cleaning, lubrication, disassembling, assembling, and repair of the equipment. The use of special tools furnished with the installation should be adequately explained.

Plans

10.9 Plans of the following equipment, where furnished, should be supplied for each ship.

Compressors — *Cooling coils*
Condensers — Heating coils
Receivers — Pumps
Brine coolers — Prime movers
Fans — *Heat interchangers*

The plans should contain sufficient information of all equipment for reordering purposes.

Diagrammatic *Piping* Layout, Coil Layout, Electric Wiring Diagram, Diagram of Pneumatic Controls, and Oil *Piping* Diagrams should also be provided. A spare parts list and a list of special tools should also be included.

Maintenance and Inspection

10.10 ***Removal of Moisture.*** Under no circumstances should alcohol or other antifreeze preparations be added to the system for purposes of overcoming freezing due to moisture. The removal of moisture from the system should be performed by use of a suitable *dehydrator* such as activated alumina or silica gel, or by means of dehydration by a suitable vacuum pump.

10.11 ***Cleaning of Condenser.*** The internal surface of *condenser* water tubes should be cleaned once every six months, or more often if found desirable. The frequency for the inspection of sea water *condensers* will depend upon service trade routes, and types of water encountered.

It is recommended that, when shutdown or stand-by equipment has been operated with harbor water, the *condensers* should be flushed out with clean sea water at the first opportunity.

10.12 ***Cleaning of Cooling Coils.*** Forced air type *cooling coils* should be inspected weekly when in use, and cleaned as often as necessary to prevent accumulation of dust or lint.

10.13 ***Brine.*** Where *brine* is used, a test of density and pH value should be conducted once each voyage and corrective treatment applied when necessary.

10.14 ***Periodic Check of System.*** A complete check of all temperatures and pressures throughout

V.C.N
27/12/80

the system should be made at least every four hours during the operation of the plant. The oil level in the *compressor* should be checked at frequent intervals and at least twice during the periodic check of temperatures and pressures.

10.15 ***Leak Testing.*** A suitable *leak detector* should be used for finding refrigerant leaks in the system. Where evidence of leakage is apparent, a check for location of such leaks should be made and repairs effected.

10.16 ***Dehydration—Freon Systems.*** After the system has been closed and tested following opening for major repairs, it should be dehydrated. This should be accomplished by evacuating the system with a suitable vacuum pump. The *compressor* should not be used for evacuating the system nor should chemical dehydration alone be used for this purpose. Evacuation should be conducted until 29.7 in. Hg vacuum (0.20 in. Hg abs. pressure) has been obtained. It should be noted that, at lower atmospheric temperatures, this process may take several days. Dehydration by evacuation should not be attempted if the ambient air temperature is below 40 F. An instrument with sufficient sensitivity should be used for determining the degree of vacuum.

59.89

10.17 ***Defrosting.*** The *cooling coils* should be defrosted, where required, at frequent intervals to permit satisfactory operation of the plant.

10.18 ***Relief Valves.*** Relief valves should not be tested on board ship under any circumstances. They should be checked to determine if the seal is intact. If the seal is broken, the relief valve should be replaced as soon as practicable.

10.19 ***Inspection of Compressors.*** The *compressors* should be opened for inspection in accordance with the requirements of the Classification Society Rules. Where a periodic inspection of the *compressor* is not required by a Classification Society, it is recommended that the *compressor* be opened for examination approximately once every two years for reciprocating and once every four years for centrifugal types. The *compressors* should be checked at frequent intervals for proper operation. A procedure for checking the operation of a reciprocating type *compressor* without dismantling is as follows:

ECKED

.5.97

10.19.1 *Testing Compressor Discharge Valve*

Pump down the *compressor* crankcase to approximately 2 psi. It may be necessary to pump down the *compressor* several times in order to vaporize the refrigerant which is mixed with the crankcase oil. If the crankcase pressure increases at a rate exceeding 3 psi per minute, it is an indication that the *compressor* discharge valve is not seating properly. If a defective discharge valve is indicated, the *compressor* should be opened for further inspection.

10.19.2 *Testing Shaft Seal*

The *compressor* should be operated at the maximum permissible *head pressure* before being shut down and the crankcase should be permitted to balance off at as high a pressure as possible (not less than 30 psi for Freon-12). The shaft seal should then be tested with a suitable *leak detector.* (It should be noted that for proper operation the seal may be designed to pass a small amount of oil, and a small leak of refrigerant may be detected. See also manufacturer's instruction book.)

10.19.3 *Compressor Operation*

Serious wear of moving parts can usually be determined by observing the operation of the *compressor.* Noisy operation not caused by refrigerant or oil slugging may be due to the wear of parts such as piston pins, connecting rod bearings, noisy valves, loose flywheel or motor pulley, misalignment on direct connected units, or improperly adjusted or worn V-belts. External components should be checked first. If the cause of the noise is not due to these components, the *compressor* should then be opened for an inspection of the internal parts.

When the *compressor* is equipped with an oil pump, the oil pressure should be observed during the operation to determine if it conforms to the manufacturer's specifications.

10.19.4 *Gaskets*

A test should be made with a suitable *leak detector* to determine if there are any refrigerant leaks around gaskets and cap screws.

10.20 ***Inspection of Instruments.*** A periodic check of all measuring instruments such as gages, industrial thermometers, dial indicating thermometers, and other measuring devices should be made with a suitable check instrument to determine their proper operation and accuracy. The frequency of inspection depends upon the service and conditions of operation but should be conducted at least once a year.

10.21 ***Inspection of Controls.*** A periodic check of all automatic controls such as low pressure cutouts, high pressure cutouts, *solenoid valves, thermostats,* and other control devices to determine their proper operation and setting should be made. The frequency of inspection depends on service and condition of operation, but should be conducted at least once a year.

10.22 ***Inspection of Spare Parts and Special Tools.*** Inspection should be made of all spare parts and special tools as to completeness and condition at least once a year. All used spare parts should be replaced promptly with new parts.

CHECKED
2008

VERIFIED
2013

SV2019

CENTRAL FOOD TECHNOLOGICAL RESEARCH INSTITUTE
LIBRARY